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Experimental study of an air-to-air, two-phase thermosiphon loop heat exchanger.

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EXPERIMENTAL STUDY OF AN AIR-TO-AIR, TWO-PHASE THERMOSIPHON LOOP HEAT EXCHANGER

BY

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TO MY WIFE, LISA

ABSTRACT

The effectiveness of an air-to-air, two-phase, coil loop thermosiphon heat recovery system using R-11 as its working fluid was experimentally investigated to determine how the performance of such systems would be affected by the hot air supply to cold air supply temperature difference, the previous thermal history of the system, and the air flow rate through the evaporator and condenser coils.

The test apparatus which could be run as either a single loop or as four identical loops in a counter flow configuration, was instrumented to measure air stream velocities and temperatures as well as the R-11 mass flow rate and inlet and outlet temperatures through each of the loops. Consideration was made for the barometric pressure, the static pressure, and the relative humidity of the air streams passing through the coils.

Upon start up of the system, it was found that a 12 - 14 C temperature difference was required to initiate boiling on the copper / R-11 interface. As the temperature difference increased to 40 C, the effectiveness of the system was found to increase and to approach a constant value. Upon cooling, the effectiveness did not decrease as rapidly and did not become zero until a temperature difference of 4 C was reached thus creating an hysteresis loop on an effectiveness versus overall temperature difference plot. This phenomenon was observed for all loop configurations and air flow rates studied.

To explain the hysteresis effect it was postulated that the number of nucleation sites grows as the overall temperature difference increases above that required for initiation and that,

upon cooling, nucleation sites will only start to be quenched whenever the overall temperature difference of the system is reduced by a magnitude equal to that required to originally initiate the boiling process {12-14 C}.

The single loop performance could be modelled by a simple exponential equation. Using the single loop test results, an iterative computer program was developed which could predict the performance of multiloop systems with good accuracy.

It was concluded that the thermosiphon system should not be designed to operate where the overall temperature difference is of the order of that required to initiate boiling.

It should also be noted that the effectiveness approaches a limit less than 1 as the number of loops approaches infinity. The value of this maximum effectiveness is dependent upon the incipient boiling temperature difference.

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1. INTRODUCTION

This study reports the findings of an experimental investigation into the performance characteristics of a prototype air-to-air two phase, thermosiphon heat recovery system. The prototype was designed for industrial applications and was tested using R-11 as the working fluid. A schematic of a two-phase thermosiphon loop heat exchanger system is shown in Figure 1.1.

Two-phase systems, like single-phase liquid forced circulation run-around systems, consist of two heat exchanger coils between which circulates a working fluid. The two-phase thermosiphon loop however, does not require power to pump the fluid from one heat exchanger to the other but instead, relies on the difference in saturation pressures between the evaporator and the condenser coupled with a gravity condensate return to circulate the working fluid. Its operation is as follows: The hot source fluid passing through the evaporator coil causes the working fluid to boil at a saturation pressure governed by the evaporator tube wall temperature. The cooler sink fluid passing through the other coil causes the working fluid to condense at a lower saturation pressure. The saturation pressure difference between the two coils causes the vapour to flow from the evaporator coil to the condenser coil. The liquid/vapour mixture leaving the evaporator tubes passes through a separator which permits the liquid to recirculate back to the evaporator and the vapour to flow through interconnecting piping to the condenser. The geometry of the system must be such that the condensate can return to the evaporator by gravity. This process essentially takes advantage of the large latent heat of vapourization which enables substantial amounts of energy to be transported per unit mass of working fluid circulated, while utilizing the high heat transfer coefficients associated with the boiling and condensing process.

For this study, the effectiveness of an air-to-air two-phase thermosiphon heat exchanger (TSHE) system using R-11 as the working fluid was measured experimentally as a function of the hot and cold air stream temperature difference, the previous thermal history of the system, and the air flow rate through the evaporator and the condenser coils. The TSHE experimental rig

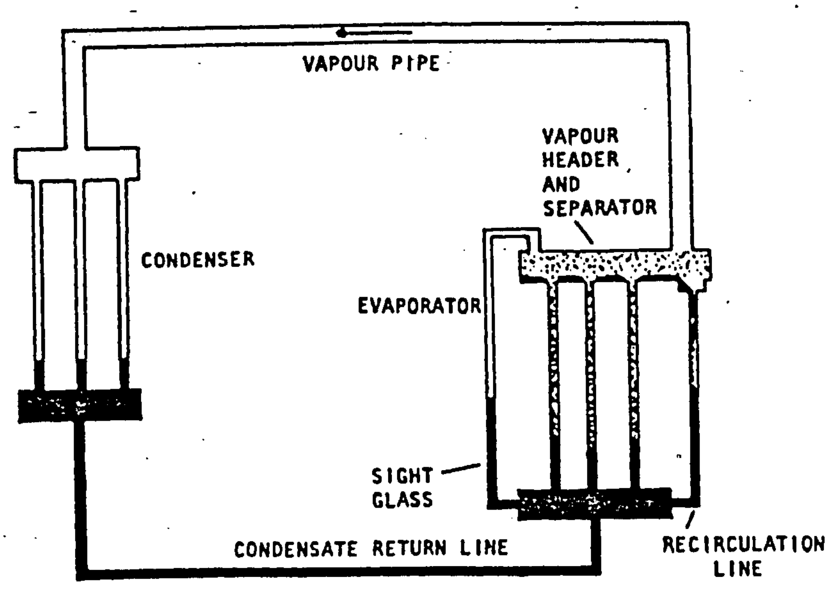

FIGURE 1.1
TWO-PHASE, SINGLE LOOP, THERMOSIPHON HEAT EXCHANGER

was arranged to study a 4 loop system configuration with each loop consisting of 2 rows of tubes in the evaporator and 2 rows of tubes in the condenser. Although not investigated here, the system could be arranged into 8 separate loops with refrigerant circulating between one row of evaporator tubes and one row of condenser tubes per loop.

The performance of a single loop, 2 row/loop system was also examined using only one loop of the 4 loop test apparatus. Data collected from these tests were later used to predict the behaviour of multi-loop systems.

The measure of performance used throughout this analysis was the 'effectiveness' of the system, and is defined as;

$$\text{Effectiveness} = \frac{\text{Qactual}}{\text{Qmaximum}}$$

where Qactual is the amount of energy {W} transferred from the evaporator to the condenser and Qmaximum is the maximum amount of energy {W} that could be transferred in the system. These energies, Qactual and Qmaximum, are determined from the difference in air stream enthalpy flux across each heat exchanger, and from the difference between the upstream evaporator and upstream condenser air inlet stream enthalpy fluxes respectively. The effects of humidity, heat loss through the ducting walls and interconnecting piping, and the air stream static and dynamic pressures have been accounted for in the calculation of these quantities.

In all tests, the effectiveness was plotted against the overall temperature difference of the system which was defined as:

$$\text{OTD} = \text{Thot} - \text{Tcold}$$

where Thot is the arithmetic average temperature (upstream of the evaporator) and Tcold is the arithmetic average temperature (upstream of the condenser).

2. LITERATURE SURVEY

Recent developments in thermosiphon technology have been largely restricted to the research performed at the University of Windsor. Previous ASHRAE grants 140 and 188 have provided financial assistance without which these projects would not have been possible.

The first loop studied some 9 years ago by Hwang{1}, and shown in Figure 2.1 , was a simple rectangular configuration constructed from 3/8 inch copper tubing. The plane on which the loop was mounted could be rotated about an axis normal to the plane and also be inclined from the vertical while water jackets on two opposite legs of the loop supplied a temperature differential across the system. Hwang studied how loop conductance was affected for changes in the temperature drop across the evaporator and condenser, the % static charge in the system and the type of refrigerant used. In summary he found that refrigerant R-11 transported more heat than that of R-113 , and that performance was optimized when the interior evaporator surface was everywhere wet and negligible flooding occured in the condenser.

These results were verified by Diciccio{2} using a modified loop design which featured glass evaporator and condenser cylinders through which 1/2 inch copper tubes were placed. Water was channelled through the copper tubing at various source and sink temperatures. The rectangluar loop configuration used in Diciccio's work could also be rotated and inclined from the vertical. Diciccio's apparatus was primarily designed for flow visualization but was equipped to measure the wall temperature

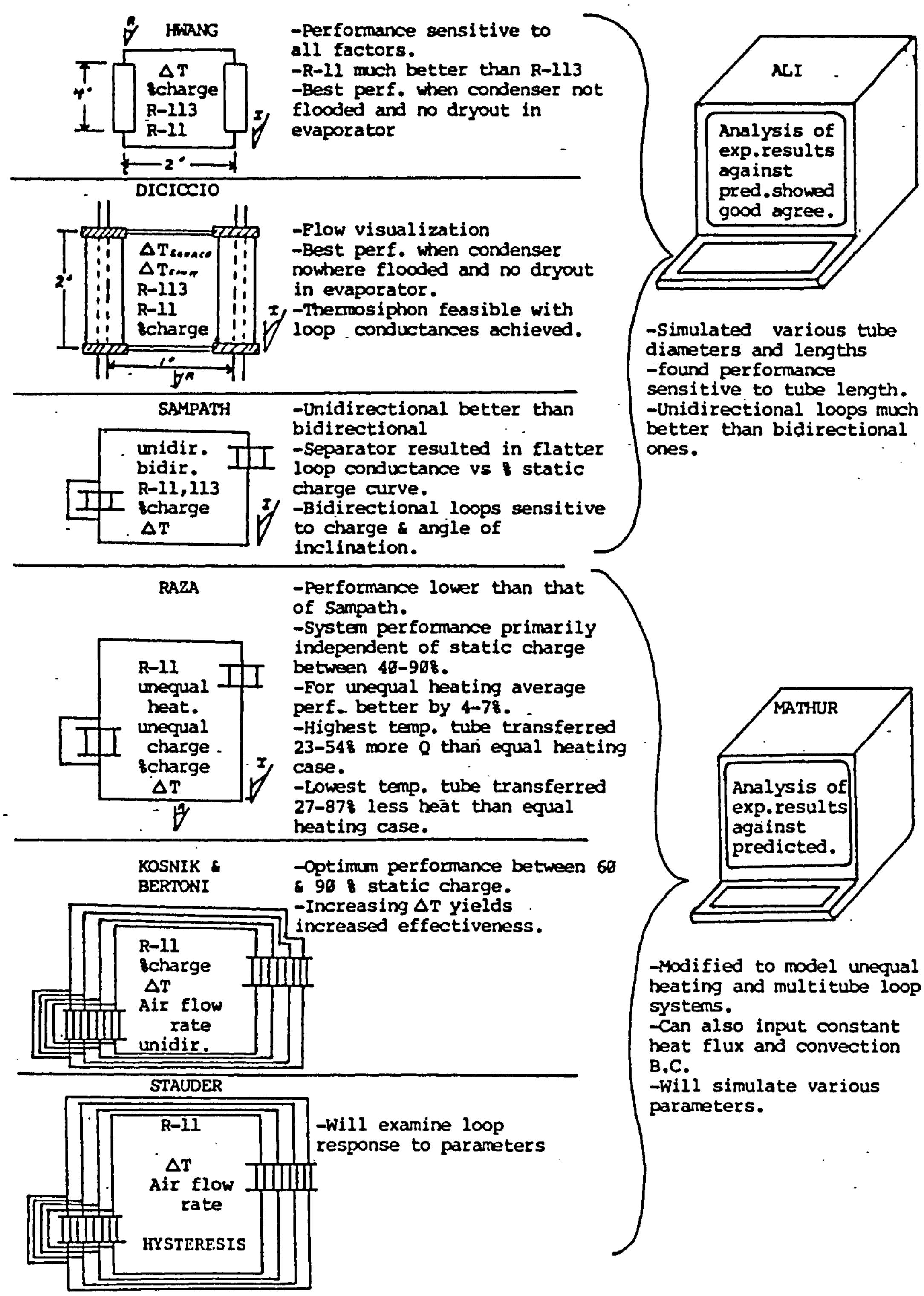

FIGURE 2.1
UNIVERSITY OF WINDSOR THERMOSIPHON LOOP RESEARCH HISTORY

and the energy transferred by the evaporator and condenser. Like Hwang, Diciccio found R-11 to achieve higher loop conductances than R-113 with optimum conditions found when there was no condenser flooding and no evaporator dryout. It was also observed that for a given loop orientation and source-sink temperature difference, optimum conductance could be achieved by varying the static charge resident in the system.

To reduce condenser flooding and evaporator dryout, a multi-tubed test apparatus was built by Sampath{3} which allowed independent positioning of separate evaporator and condenser units. A liquid/vapour separator and condensate return tube was also installed at the evaporator outlet header to prevent liquid carryover from the evaporator into the condenser, while suppressing the onset of dryout by allowing the evaporator to run with a greater charge. In addition to the refrigerant type, the amount of static charge , and the temperature drop across the system, Sampath also looked at the effect of unidirectional and bidirectional configurations on the loop conductance. Bidirectional loops are oriented so that working fluid is resident in both heat exchangers under static conditions and as such, the refrigerant flow may travel in either direction through the loop depending upon which heat exchanger was used as the evaporator and which was used as the condenser. Unidirectional loops , or thermal diodes, will only permit the flow of working fluid in one direction since under static conditions, one heat exchanger is wet with working fluid and the other is elevated and dry. With this arrangement only the heat exchanger which contains the liquid working fluid can be used as the evaporator.

Sampath found that the unidirectional loops achieved higher loop conductances than the bidirectional loops since condenser flooding was seen to occur in the later, and that refrigerant R-11 will transport more energy from the evaporator to the condenser in the range of temperatures studied than will R-113. Sampath also found that for unidirectional loops, the loop conductance was highest when the condenser was inclined 15 to 75 degrees from the vertical. The loop conductance did not vary substantially over this range of condenser inclination angles.

The liquid/vapour separator installed on Sampath's apparatus was observed to permit lower temperature difference operation by reducing the amount of liquid carryover into the condenser and to suppress dryout in the evaporator by supplying additional refrigerant to the evaporator's liquid header.

Utilizing correlations available in the literature, Ali{4} developed a computer program to predict the behaviour of such thermosiphon loops. The simulated results showed good agreement with the experimental data of the first three studies and was therfore used to predict how the performance would vary for other tube diameters and lengths. In general, Ali found the maximum loop conductance to decrease for increased evaporator tube diameters, and that the maximum loop conductance for increased evaporator tube diameters is found by increasing the tube length.

Raza{5} using the same apparatus as Sampath, studied the effect of unequal evaporator tube heating and the effect of unequal charge distribution in the evaporator tubes. By individually water jacketing each tube of the evaporator and condenser, a mean temperature difference between the tubes could be created.

Raza found that for charge variations of 3 to 13% between adjacent evaporator tube rows caused by inclining a coil, the loop conductance was negligibly affected.

The heat transfer rate was found to increase slightly {4% to 7%} for unequal heating of a 3 row evaporator where the temperature differences between tube rows was as much as +- 40% of the average source to evaporator saturation temperature difference. This comparison was made against the performance where all 3 tube rows were subjected to the same source fluid temperature as the middle tube row in the unequal heating tests.

On average, 23%-54% more heat was transfered in the evaporator tube subjected to the highest source fluid temperature and 27% -87% less heat was transferred by the tube subjected to the lowest source fluid temperature than would be expected if all the tubes were subjected to the same high source fluid temperature. 9% more heat was transferred in the evaporator tube subjected to the average source fluid temperature than would also be expected for equally heated tubes.

Raza also noted that the system performance did not vary appreciably for static charges of between 40 and 90%.

Experiments involving a commercial size thermosiphon heat exchanger system by Kosnik and Bertoni{6} was the most recent study carried out at the University of Windsor. Their system utilized two commercially available alumimum finned, air-to-air, 3 by 1 foot heat exchanger coils between which circulated a working fluid of R-11. The evaporator coil consisted of 8 separate vertical rows of tubes with independent horizontal vapour and liquid headers for each. All 8 rows of tubes were fitted with a vapour/liquid separator and liquid return line as suggested by Sampath. The condenser coil was inclined at 45 degrees to the vertical and also consisted of 8 separate rows of tubes with independent horizontal vapour and liquid headers. Each evaporator vapour and liquid header was connected via 1 inch copper pipe and 3/8 inch flexible tubing respectively, to a corresponding condenser vapour and liquid header. The test apparatus could be arranged to study 8, 4, 2, or 1 loop systems configurations. For their study, a 4 loop, 2 rows of tubes per loop system configuration was used to observe the effect on performance of static charge level, overall system temperature difference, and coil air face velocity. Unlike the previous studies, the measure of performance used in Kosnik and Bertoni's work was the systems 'effectiveness'. Kosnik and Bertoni found that the 4 loop system operates best between static evaporator charges of 60 and 90% with optimum performance achieved near 80%. They also observed that when the system overall temperature difference increased, effectiveness increased. Also, imposed changes in the coil heat exchanger face velocity were seen to shift the performance curve (effectiveness versus overall temperature difference) up for decreasing face velocities and down, for increasing face velocities.

Mathur{7}, using empirical relations found in the literature, has extensively modified Ali's original computer simulation program to predict the behaviour of multi-row, multi-loop, system configurations. Upon comparison, Mathur has found good agreement between his computer generated results and the data experimentally determined by Raza and by Kosnik and Bertoni.

Work performed outside the University of Windsor and available in the literature concerning boiling phenomena have indicated that a wall surface temperature substantially above the local saturation temperature is required to initiate boiling. One of the first reports of this phenomena is by Abdelmessih et al. {8} who experimentally observed this boiling phenomena in an electrically heated, horizontal , stainless steel tube test section using Trichlorofluoromethane {R-11} in forced flow. A wall superheat of 25 to 30 degrees F (13.8 to 16.7 degrees C) above the local saturation temperature was found necessary to initiate boiling in their test facility. Once local boiling was initiated, the wall superheat dropped substantially. This nucleate boiling could be sustained even when the wall heat flux was reduced to the point where the tube wall temperatures fell to within a few degrees of the local saturation temperature. Further research into this phenomenon has been reported by a variety of investigators including Joudi and James{9} , who observed that wall superheats of approximately 16 degrees C for R-113 and 19 degrees C for Methonol were required to initiate boiling. Their tests were performed using a flat horizontal stainless steel surface of known roughness at atmospheric pressure.

A study by Marto and Rohsenow{10} involving commercial grade sodium in pool boiling from a horizontal disk reported superheat temperatures as high as 135 degrees F (75 degrees C) depending upon the finish of the disk surface. They concluded that " the wall superheat required to initiate nucleate boiling of sodium is significantly greater than that required to sustain nucleate boiling". Similarly, Turton {11} reported that the incipient boiling wall superheat temperature necessary for R-11 in a stainless steel, electrically heated, tube test section under increased gravity of 2.5g and a pressure of 88 lbsf/in2 was found to be as high as 120 degrees F (66.7 degrees C).

Recent work by Robertson and Clarke {12,13} have established similar conclusions using liquid Nitrogen in electricaly heated, serrated aluminum plate-fin heat exchangers. They found a wall superheat of 5 Kelvin degrees was necessary to initiate nucleate boiling with liquid Nitrogen. Robertson and Clarke also incrementally cycled the inlet temperature of the liquid Nitrogen to observe if established nucleation sites would remain upon cooling and reheating. They concluded that the "sites on the walls from which the bubbles grew were inactivated when only liquid Nitrogen flowed past them". This suggests that the nucleation sites were quenched upon substantial cooling.

It would appear from the research available in the literature that a delay in the onset of nucleate boiling of refrigerant R-11 could be expected in the TSHE and as such, might adversely affect the performance of these systems. It was thus the intention of this study to determine if a delay in the onset of nucleate boiling was present and how this phenomenon might effect the performance of the same 4 loop, 2 row of tubes per loop prototype commercial air-to-air thermosiphon heat exchanger test apparatus previously utilized by Kosnik and Bertoni.

3. APPARATUS

The test facility shown in Figure 3.1, consists of two 0.9 by 0.3 meter {3 x 1 foot} air to fluid finned heat exchanger coils interconnected by tubing. The evaporator contains 8 separate, 0.9 meter long, vertical, staggered , rows of twelve nominal 3/8 inch copper tubes with horizontal liquid and vapour headers for each row. The condenser, which was elevated above the evaporator such that no condenser flooding occurred, consists of 8 separate, 0.3 meter long, staggered, rows of thirty six nomimal 3/8 inch copper tubes with horizontal liquid and vapour headers for each row. The condenser was mounted with its tubes inclined at 45 degrees to the vertical. Both heat exchanger coils have tubes spaced at 2.54 cm {1 in} centres and are fitted with 4.7 wavy aluminum fins/cm {12/in} and are 17.8 cm {7 in} deep.. The eight separate evaporator vapour headers are each connected via nominal 1 inch diameter copper pipe in a counterflow arrangement with the eight separate condenser vapour headers. Likewise, the condensate returns from the condenser liquid headers through 9.5 mm {3/8' in} flexible Nyloseal tubing to the refrigerant flow meters located just upstream of the evaporator liquid headers. The liquid and the vapour headers of each row of evaporator tubes were interconnected by an external sight glass, which provided both a measure of the pressure drop between the headers and an indication of the dynamic charge in each row of tubes, and by a recirculation tube which permitted liquid from the separator to return to each evaporator liquid header.

For this study, the 8 independent loops were combined in pairs on both the liquid and vapour side of the evaporator to produce 4 independent loops, with each loop consisting of two rows of evaporator tubes and two rows of condenser tubes. Ducting was designed to supply equal air mass flow rates through both coils while utilizing a 0 to 59 KW {200,000 Btu/hr} variable firing rate, variable fan speed, forced air gas furnace. Maximum and minimum air temperatures achievable in the ducting were 70 degrees C {158 F} and outside ambient air respectively for face velocities of between 1 and 3.6 m/s {200 to 700 fpm}.

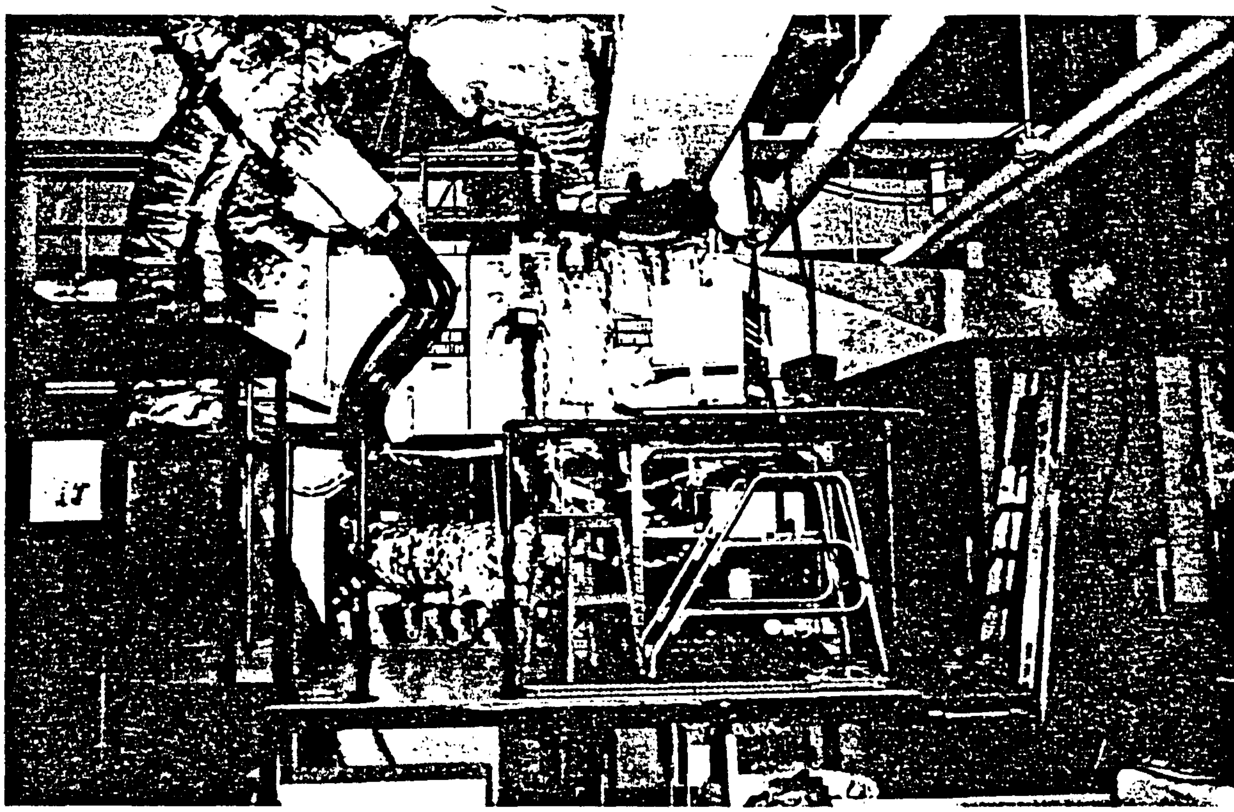

FIGURE 3.1
THERMOSIPHON HEAT EXCHANGER TEST APPARATUS FACILITY

A variable area mixing chamber near the cold air inlet permitted warm exhaust air to be diverted into the cold air supply duct and thus gave some flexibility in the cold supply air temperature. This arrangement as well as the location of three commercial grade air filters within the ducting can be seen in Figure 3.2. Appendix A contains the mixing chamber dimensions and outlines the shortcomings of this arrangement.

Gilmore tappered tube and float flowmeters were installed into each of the refrigerant loops immediately upstream of the evaporator to give both an indication of the R-11 evaporation rate and dynamic stability in each of the loops. The flowmeters were originally purchased with a single glass float and steel float in each which the manufacturer claimed would be capable of measuring the flowrates expected in the TSHE test apparatus. Upon receiving the flowmeters, a calibration check was first performed which found the maximum measurable flow rate through the flowmeters to be substantially less than suggested by the manufacturer. This prompted the design of a second float which was capable of indicating flowrates greater than that of the floats supplied by the manufacturer. All 8 flowmeters were then re-calibrated with both a glass and designed float in each tube prior to installation. The use of the two floats allowed the flowmeters to be used over an expanded range of volume flow rates. A detailed description of the installation and calibration procedure can be found in Appendix B.

System temperatures were obtained using 90 copper - constantan thermocouples throughout the apparatus. A grid of 18 thermocouples were placed in the air flow both upstream and downstream of the evaporator and condenser with two additional thermocouple installed into each of the 8 loops to measure the refrigerant temperature at entrance to and exit from the evaporator. Thermocouples were used to measure the ambient room air temperature and the outside condenser vapour header temperature. This latter thermocouple could be easily moved to measure any other system temperature desired. A thermocouple location index is supplied in Appendix C.

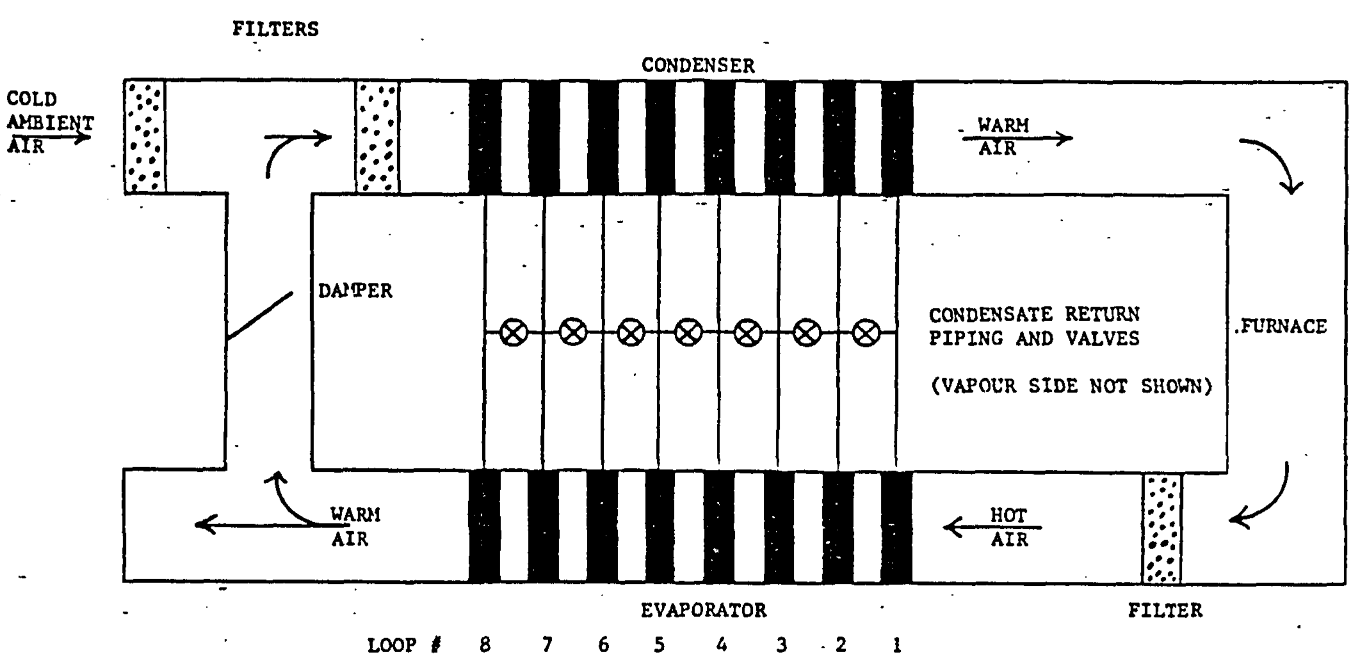

FIGURE 3.2
SCHEMATIC REPRESENTATION OF TSHE TEST FACILITY APPARATUS

4. INSTRUMENTATION

The output from the 90 copper-constantan thermocouples were fed directly into the scanner boards of a 2240B Fluke Data Logger which was interfaced with a 64 K Apple computer equipped with a Z80 card. This computer controlled data acquisition system, shown in Figure 4.1 below, was used to scan and store the output of all the thermocouples. The temperature data, together with manually inputted values for air pressure, dew point, air velocity, and refrigerant flow rate, were used to calculate the air mass flow rate and the enthalpy flow rate upstream and downstream of both coils, the refrigerant mass flow rates, the energy transfer rate and the effectiveness of the system.

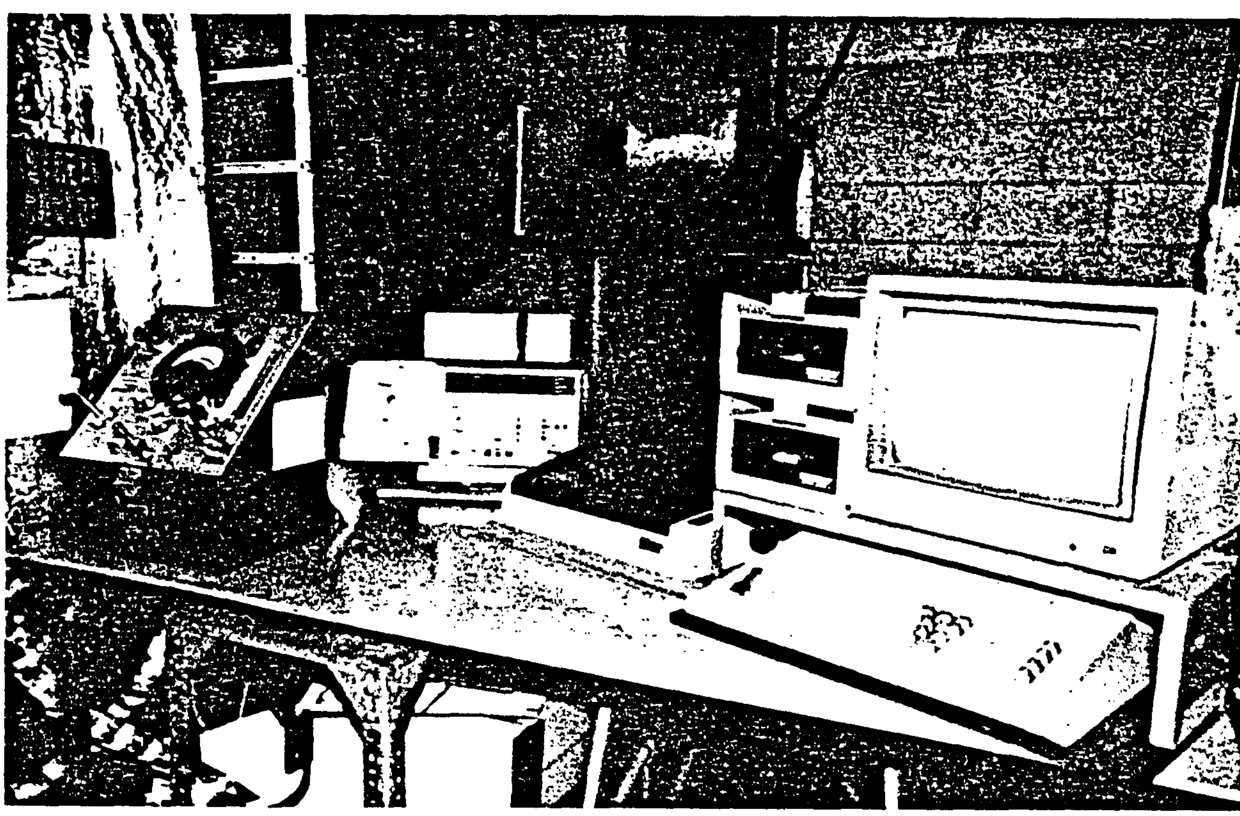

FIGURE 4.1
DATA ACQUISITION SYSTEM CONSISTING OF 64K APPLE COMPUTER AND FLUKE 2240B DATA LOGGER.

The dew point temperature of the duct supply air stream was obtained using an Alnor Dew Pointer which extracted a small sample of air from the air flow upstream of the condenser for analysis.

In order to measure the air stream velocity within the duct, an Alnor flow through Velometer was used with a specially constructed, 2 port probe. Both the velometer and probe were calibrated together against a manometer prior to use. A detailed description of the calibration procedure and probe configuration can be found in Appendix D.

5. SYSTEM PREPARATION

5.1 RESERVOIR SYSTEM

To ensure that the refrigerant R-11 was reusable after being drained from the system when effecting repairs, a permanently connected reservoir was constructed which enabled the working fluid to be injected or extracted from the system without exposure to the surrounding air.

The reservoir system shown in Figure 5.1.1 , consisted of a standard 100 Kg capacity, refrigerant drum which was mounted on a vertical, sliding platform that could be raised or lowered to any desired position relative to the TSHE apparatus by means of a winch and pulley. The reservior tank was equipped with two valves, one for the liquid flow and one for the vapour, so that

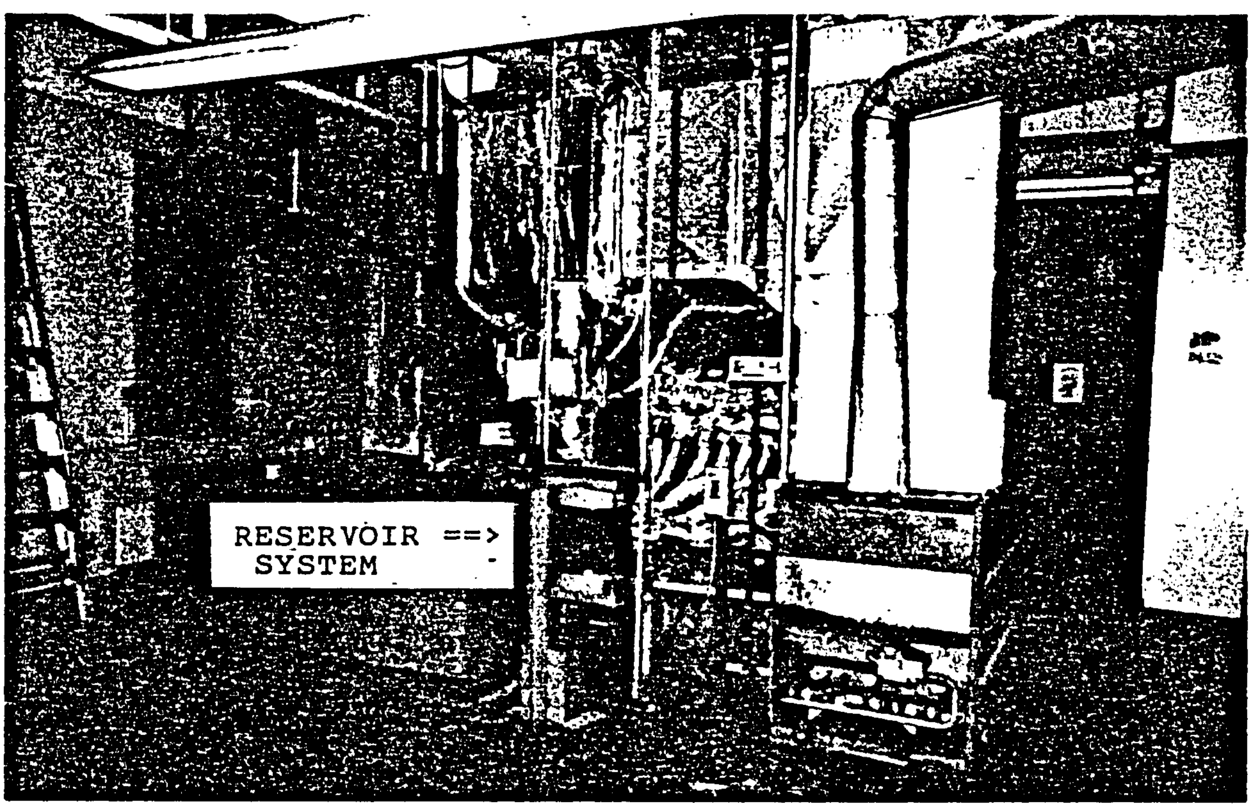

FIGURE 5.1.1
THERMOSIPHON TEST APPARATUS RESERVOIR SYSTEM

the tank could be completely sealed from a drained TSHE system. Figure 5.1.2 shows a sketch of the reservior system and how it was integrated with the TSHE test apparatus.

To raise the static refrigerant level in the evaporator, the reservoir tank was first elevated to a height above the evaporator using the winch and pulley arrangement. The

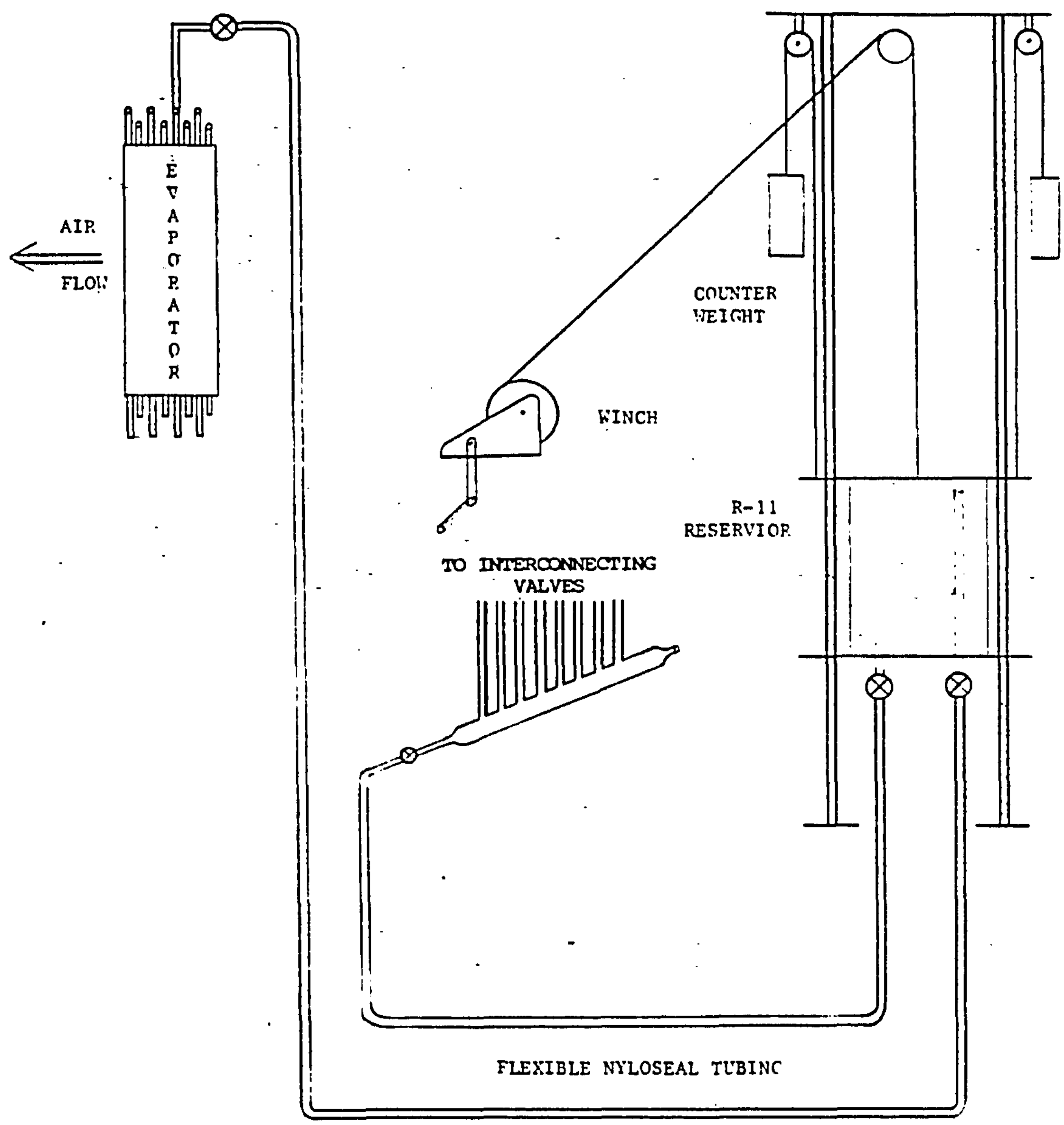

FIGURE 5.1.2
SCHEMATIC OF RESERVOIR SYSTEM INSTALLED ON TSHE APPARATUS

interconnecting valves were then opened causing the R-11 to flow from the tank to the room temperature TSHE apparatus because of the pressure differential now created between the tank and the system.

To reduce the amount of static charge resident in the evaporator, the reservoir tank was lowered to an elevation below the evaporator thereby allowing the refrigerant to flow from the TSHE apparatus to the tank by gravity.

5.2 LEAKAGE TEST

Two methods of leak detection were used after repairs had been completed on the TSHE apparatus. The first involved monitoring the pressure in the loops after injecting compressed Nitrogen gas several hours earlier. If a leak was detected by a decrease in the systems pressure , a soap/water solution was used on suspected areas to help pin-point the exact location of the perforation and thus, effect repairs before charging with R-11.

The second method used to detect system leaks utilized a commercial Halogen detector. This device emitted a `beep` which increases in frequency when higher concentrations of halogens are encountered. The Halogen detector was often used in conjunction with the soap/water solution both before and after repairs had been made while the refrigerant was resident in the system.

5.3 VACUUM PUMP

The vacuum pump, used to extract non-condensible gases {air} from the system before charging with R-11 was equipped with two liquid-gas separators immersed in liquid Nitrogen. These were used in series immediately before the pump to condense any refrigerant vapour that might be drawn out of the TSHE system. A vacuum of approximately 29 inches of mercury was achievable when using this arrangement for several hours.

5.4 CHARGING PROCEDURE

Throughout this research the following charging procedure was developed and adhered to. For future studies this method is strongly recommended.

1. After all repairs have been made, perform a leak check with compressed Nitrogen.

2. Once the TSHE apparatus is satisfactorily leak free, evacuate the non-condensible gases from the loops through the liquid Nitrogen separators by using the vacuum pump . Connect the vacuum pump to the schrader valve located on the right hand side of the charging header. The charging header is located below the interconnecting loop condensate valves on the condensate return side. Be sure to open the necessary valves to allow the non-condensible gases to be drawn out. To help ensure gas free loops, allow the pump to run for at least 6 hours. Once completed, let the pump continue to run while disconnecting it from the system.

3. Raise the R-11 reservoir container to a height above the evaporator and open the charging header valve, both reservoir container valves, and the evaporator vapour header valve. A completely empty TSHE system will take approximately 4 minutes to reach a static charge level of 29 inches above the bottom of the evaporator which corresponds to filling the evaporator 80% full

of refrigerant.

4. Once the desired level of charge is obtained, close the charging header valve, both reservoir container valves, and the evaporator vapor header valve and set the interconnecting loop valves to the required configuration.

5. To ensure that the system does not contain some form of non-condensible gas, it is strongly suggested that the following Purging Procedure be next initiated.

5.5 PURGING PROCEDURE

To effectively purge the TSHE of non-condensible gases, the following procedure and computer program was developed.

1. Close all R-11 interconnecting valves on the thermosiphon except those between rows of the same loop.

2. Turn on furnace burner switch and open fully all 5 gas burner valves.

3. Check to ensure furnace exhaust is rising up flue stack. If not, open windows to outside air and shut door to corridor.

4. Turn on fan and set to low shaft speed with strobe {approx. 1150 rpm}

5. Place crushed ice generously around the purging headers which are located just below the condenser (do not use dry ice).

6. Set the damper to position #5 {0% open}

7. Let the system run for approximately 1/2 hour noting which loops have a reduced R-11 volume flow rate through the flowmeters.

8. Check the vapour and liquid temperatures by using the "PURGE" program to determine if purging can be started.

9. If `OK` to purge, purge at the purging headers using short bursts . Purge only those rows which showed a reduced volume flow rate in the flowmeter. DO NOT PURGE EXTENSIVELY.

10. Once purging is completed, turn off furnace burner switch and fan switch together. Check to see if purging headers require more ice.

11. Let system cool for approximately 20 minutes. (this allows the non-condensible gases to migrate to the purging header)

12. Re-start furnace burners and fan.

13. Repeat steps 7 to 12 at least 3 times to ensure the adequate purging of non-condensible gases from the system.

5.6 DISTILLATION UNIT

Throughout this study, refrigerant R-11 was considered `contaminated` and not suitable for re-use whenever exposure to the atmospheric air occurred or when the refrigerant was seen to absorb oils and/or particles. To purify contaminated refrigerant for later re-use, a distillation apparatus was constructed and is shown schematically in Figure 5.6.1 . This apparatus gently heated the contaminated refrigerant to evaporate pure R-11 vapour which was channelled to a condenser where the condensate would be drained into a clean, sealed tank. Although this process worked fairly well, the lack of necessary control valves and instrumentation on the apparatus made balancing of the pressures throughout the unit difficult. It is suggested that further refinement of this apparatus be made to improve the distilled refrigerant quality and ensure operator safety.

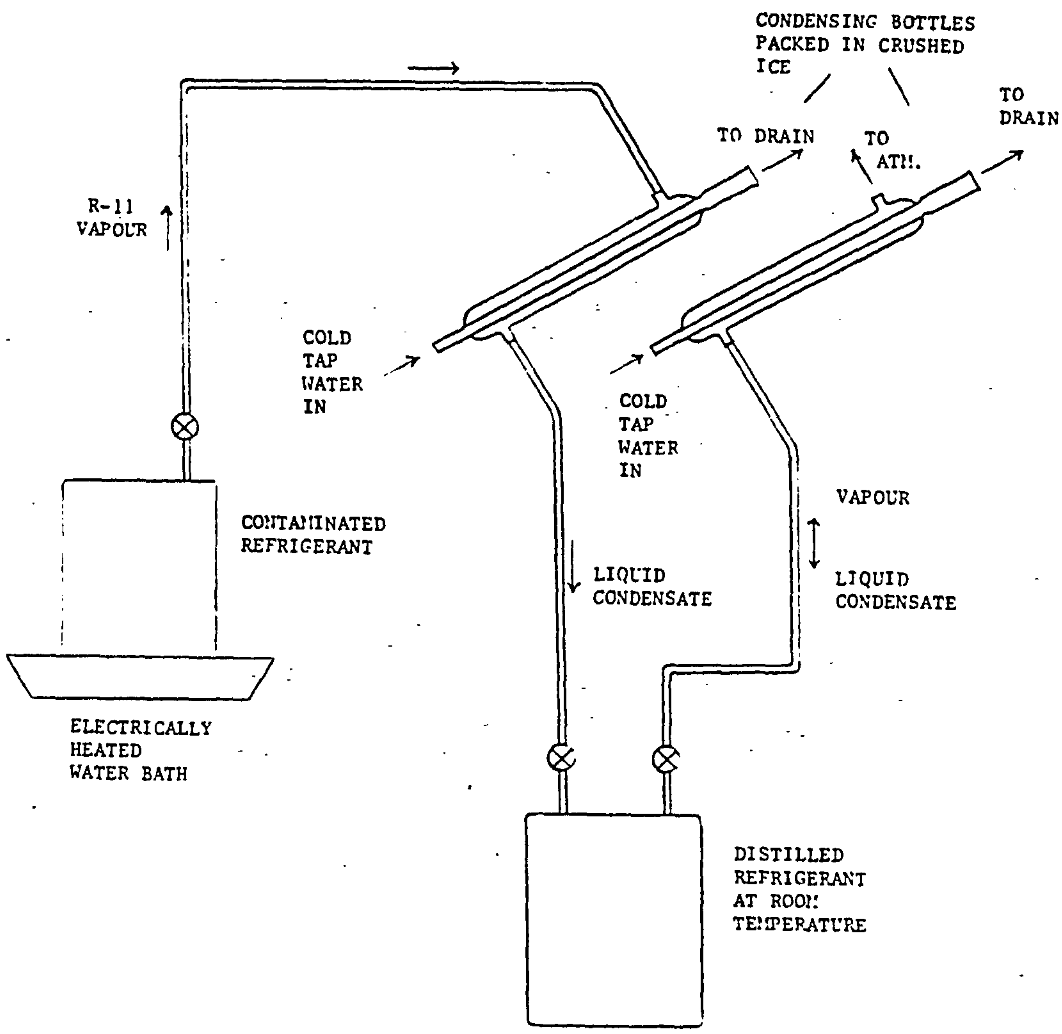

FIGURE 5.6.1
SCHEMATIC OF R-11 DISTILLATION APPARATUS

6. VELOCITY AND PRESSURE MEASUREMENTS

In order to calculate the effectiveness of the TSHE system the energy transfer rate between the evaporator and condenser, and the maximum possible energy transfer rate for the system per unit time must be determined. To obtain these quantities, the product of the mass flow rate and enthalpy of air was calculated for each of the 4 measurement stations on the test apparatus. The measurement stations are situated before and after both the condenser and evaporator heat exchanger coils.

The enthalpy of the air at each of the measurement stations was calculated using Equation 30, Page 5.3 of the 1981 ASHRAE Fundamentals Handbook. The humidity ratio and the dry bulb temperature are required for this calculation.

To determine the mass flow rate of the air at any one measurement station, the equation:

$$\dot{m} = \rho\ V\ A \qquad \text{Eqn. 6.1}$$

was used where ρ is the density of dry air, V is the velocity of the air, and A is the cross sectional area of the duct.

The density of dry air used in this equation was calculated from the ideal gas law equation:

$$\rho = \frac{(Pb - Ps + Pst)}{(R)(T)} \qquad \text{Eqn. 6.2}$$

where Pb is the outside barometric pressure, Ps is the partial pressure of water vapour, Pst is the static gage pressure within the duct, R is the universal gas constant, and T is the temperature of the air. The barometric pressure Pb, was obtained using a standard mercury barometer and the partial pressure of the water vapour Ps, was calculated using the dew point temperature with equation 4, Page 5.2 of the 1981 ASHRAE Fundamentals Handbook. To determine the static pressure created in the duct, single static pressure readings were taken with a manometer and pitot tube probe at each of the 4 measurement stations. In order to ensure correct static pressure levels for the different flow conditions imposed, static pressure readings were taken for each of the 3 fan speeds and 5 damper positions used. In total, 60 static pressure readings were obtained. An explanation of the static pressure test and a summary of the values measured can be found in Appendix A.

The temperature required for the calculation of the density of the duct air in equation 6.2 was determined using 18 copper-constantan thermocouples at each of the 4 measurement stations. The 3 by 1 foot inside duct area was divided into a grid consisting of 18 equal area rectangles in which the thermocouples were mounted at the center of each. A minimum number of 16 rectangles no greater than 6 inches between centers is recommended by ASHRAE {Pg. 13.15, 1981 Fundamentals Handbook}. Figure 6.1 is a sketch of a section of ducting in the test facility and shows how the duct area was divided at the measurement stations.

To complete the calculation of the mass flow rate(Eqn. 6.1), the velocity of the air at each of the 4 measurement stations was required. Velocity measurements were obtained using the Alnor

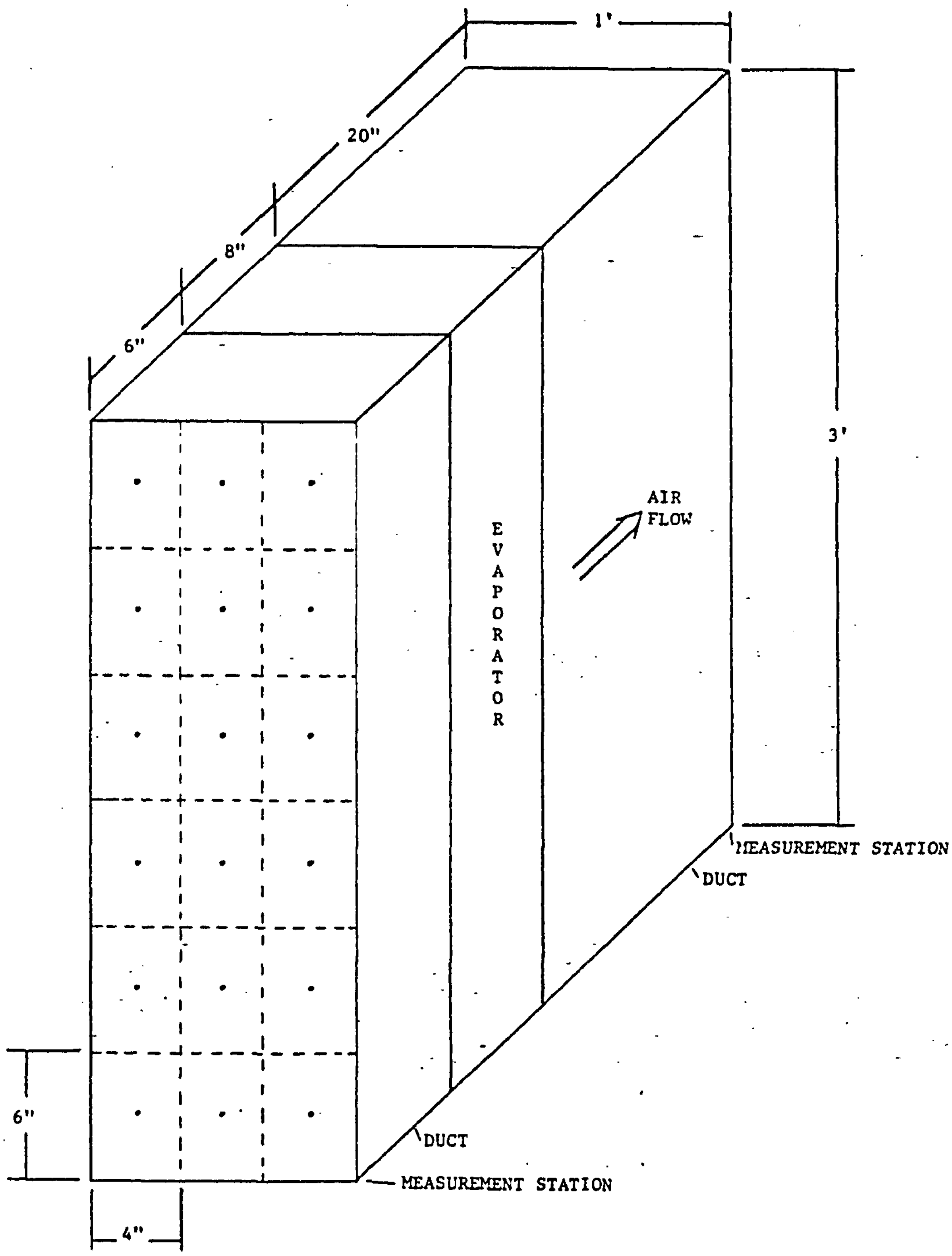

FIGURE 6.1
EQUAL AREA DUCT GRID USED FOR VELOCITY AND TEMPERATURE MEASUREMENTS AT EACH MEASUREMENT STATION

Velometer and Probe which was systematically placed using a traverse arrangement, at the center of the same 18 equal area rectangles used for the temperature measurements.

In order to ensure that the velocity measurements obtained for each equal area rectangle were representative of the average velocity through the rectangle, a detailed distribution of the velocity along a single traverse through the center of each of the measurement stations was carried out. From these detailed distributions, it was concluded that the velocity measurement obtained from the center of the equal area rectangles closest to the duct walls, did not adequately represent the average velocity through the rectangular area. A correction to the velocity profile was then introduced in order to more closely estimate the velocity distribution in those rectangles adjacent to the ducting wall.

Since the velocity measurements were taken with no heat applied by the furnace, the temperature and thus density of the air was constant throughout the ducting. This suggests that for a given air mass flow rate through the system, the average air velocity at each of the equal area measurement stations must be the same.

Upon checking the corrected velocity distributions , a small discrepancy in the average velocities between the 4 measurement stations was found. To correct for this, the averaged velocity of all 4 measurement stations was used to apply a correction to the distributions.

All 4 profiles were then individually normalized using a selected velocity reading from near the center of each profile. This normalized profile was incorporated into the data acquisition program to enable generation of the 4 measurement station velocity distributions when multiplied by manually inputted reference velometer readings obtained at the time of a test.

Prior to the performance testing of the thermosiphon system, it was determined that fan speed and damper position will appreciably change the velocity distributions at each of the 4 measurement stations.

To correct for the change in the velocity distributions due to fan speed, a set of 4 normalized velocity distributions were developed for each of the 3 different fan speed used in this study. All 12 normalized distributions were then incorporated into the computer data acquisition program for later use with the reference velometer readings.

Upon examining the change in the velocity distributions for the various damper positions available, it was found that only the distribution located upstream of the condenser at a fan speed of 2300 rpm was significantly affected. To correct the upstream condenser distribution when using the 2300 rpm fan speed, 5 more distributions were measured corresponding to the 5 different damper positions available. A correction to the velocity distribution was thus developed to account for the damper position and was also included into the data acquisition program.

Upon completion of the performance testing, the 12 velocity distributions were again checked and were determined to be within experimental error. It was though that the velocity distribution in the system might be appreciably affected by the cleanliness of the filters. Fortunately, this was proven to be unjustified.

Appendix A contains the original velocity distributions measured, a brief description of how the normalized distributions were calculated, the velocity distributions obtained for the various damper positions, and the data obtained for the duct static pressure measurements for different damper positions as well as a brief description of the damper arrangement.

7. EXPERIMENTAL PROCEDURE

7.1 TESTING PROCEDURE

The following series of steps outline the testing procedure used in this study to determine the performance characteristics of the TSHE system.

1. Ensure that the system is at room temperature and that the correct amount of refrigerant is resident in the evaporator. If so, record the liquid level; if not, add or remove charge as outlined in section 5.1 .

2. Turn fan switch to 'ON' and using the strobe, set the fan shaft speed to the desired rate. Check to ensure that the furnace exhaust is rising up the flue stack. If not, open windows to outside air and shut door to corridor.

3. Knowing the overall temperature difference required for the test, set all 5 furnace burner valves so that this temperature difference is achieved.

4. 'Boot up' and run TSHR program and follow the 'RUN' instructions.

5. Fix damper position to give the desired air temperature upstream of the condenser.

6. Obtain the dewpoint temperature and barometric pressure. Enter these into the TSHR program when so promted by the program.

7. After approx. 1 hour, begin test by taking all 4 duct air velocity reference readings and input them into the computer.

8. Following the instructions from the computer for scanning with the Fluke Data Logger, initiate the temperature scan. Obtain flowmeter readings while the data logger is scanning the system temperatures.

9. Input the flowmeter readings and wait for program to complete execution { approx. 1 min}.

10. Save test data to disk.

11. Obtain hard copy of test results from printer.

12. Increase or decrease all 5 furnace burner gas valves to achieve the next overall temperature difference required.

13. Repeat steps 5 through 12 for the temperature range desired.

An average time of 10 minutes was required to complete one test run. A typical data sheet used to record the manually read data and the computer output results can be found in Appendix E.

7.2 TESTS CONDUCTED

Table 7.2.1 summarizes the test sequences run and shows the system configuration parameters for each sequence. A policy of completing a test sequence once started was strictly adhered to and often resulted in tests which required several hours to complete.

In the work done by Raza and Kosnik and Bertoni prior to this study, it was determined that the performance of TSHE systems is fairly insensitive to refrigerant static charge levels of between 60 and 90 % of the evaporator full condition. In the data collected by Kosnik and Bertoni, the highest levels of effectiveness were achieved for static charges of approximately 80% full. For this reason, the static liquid level in the evaporator tube sight glass was maintained at a point 29 inches from the bottom of the evaporator finned tube section. This

REVISED THERMOSIPHON TEST SEQUENCE[1]
F.A.STAUDER
1/12/84

VARIABLES	SEQUENCE LETTER \ NUMBER[2]	1	2	3	4	5	6	7	8	9	10	11	12	13
FAN SPEED (RPM)		1700	1700	1150	2300	1150	2300	1700	1700	1700	1700	1700	1700	1700
TEST TYPE[3]	---	IF	DF	DF	$\boxed{\text{DF}}$	$\boxed{\text{H}}$	$\boxed{\text{H}}$	$\boxed{\text{H}}$	C	$\boxed{\text{F}}$	DF	DF	DF	$\boxed{\text{H}}$
	A	ST	C		DF	$\boxed{\text{H}}$	H	$\boxed{\text{H}}$						H
	B					$\boxed{\text{H}}$		H						
	C					H								
	D					IF								
# OF LOOPS WITH R-11		4	4	4	4	4	4	4	4	4/3 2/1	1	1	1	1
LOOP WITH R-11 (A-ALL)		A	A	A	A	A	A	A	A	A 2/3/4 3/4 4	4	2	1	1
# OF SIGHT GLASSES/ ROW		0	0	0	0	1	1	1	1	1	1	1	1	1

1 - EACH TEST IS PERFORMED FOR A NUMBER OF DIFFERENT OVERALL TEMP. DIFFERENCES
- STATIC CHARGE OF APPROX. 80%
- SYSTEM CONFIGURATION: FOUR LOOPS WITH 2 ROWS/LOOP, 1 RECIRCULATION TUBE/ROW

2

SEQUENCE #	SEQUENCE LETTER	TEST LETTER	+	DESCRIPTION
7	A	C	+	SEQUENCE 7A, TEST RUN 'C'
5	---	F	+	SEQUENCE 5, TEST RUN 'F'

3

SYMBOL	:	DESCRIPTION
H	:	HYSTERESIS TEST, HEATING AND COOLING SYSTEM ONCE.
DF	:	DECREASING FLUX TEST, OBSERVE PERFORMANCE DURING COOLING OF SYSTEM
IF	:	INCREASING FLUX TEST, OBSERVE PERFORMANCE DURING HEATING OF SYSTEM
C	:	CYCLIC TEST, REPETATIVE HEATING AND COOLING OF SYSTEM BETWEEN FIXED TEMP LIMITS
F	:	FIXED FURNACE GAS SETTINGS TEST, DRAINED LOOPS ONE AT A TIME WITH FURNACE SETTINGS FIXED
ST	:	STEADY STATE HEATING TIME
$\square$	:	TEST NOT INCLUDED IN STUDY DUE TO LARGE EXPER. ERROR

TABLE 7.2.1
TABLE OF TESTS CONDUCTED

corresponds to a static charge of approximately 80%. This amount of charge was used in all the tests conducted throughout this study.

The purpose of this investigation was to examine the steady state performance behaviour of the TSHE system for various operating conditions . To investigate how quickly the system achieves steady state conditions, test sequence 1A was carried out in which data was collected as often as possible once the furnace had been turned on with the 5 gas supply valves open fully, and the apparatus fan speed set to 1700 rpm {2.2 m/s coil face velocity}. From these results shown in Figure 7.2.1, it was

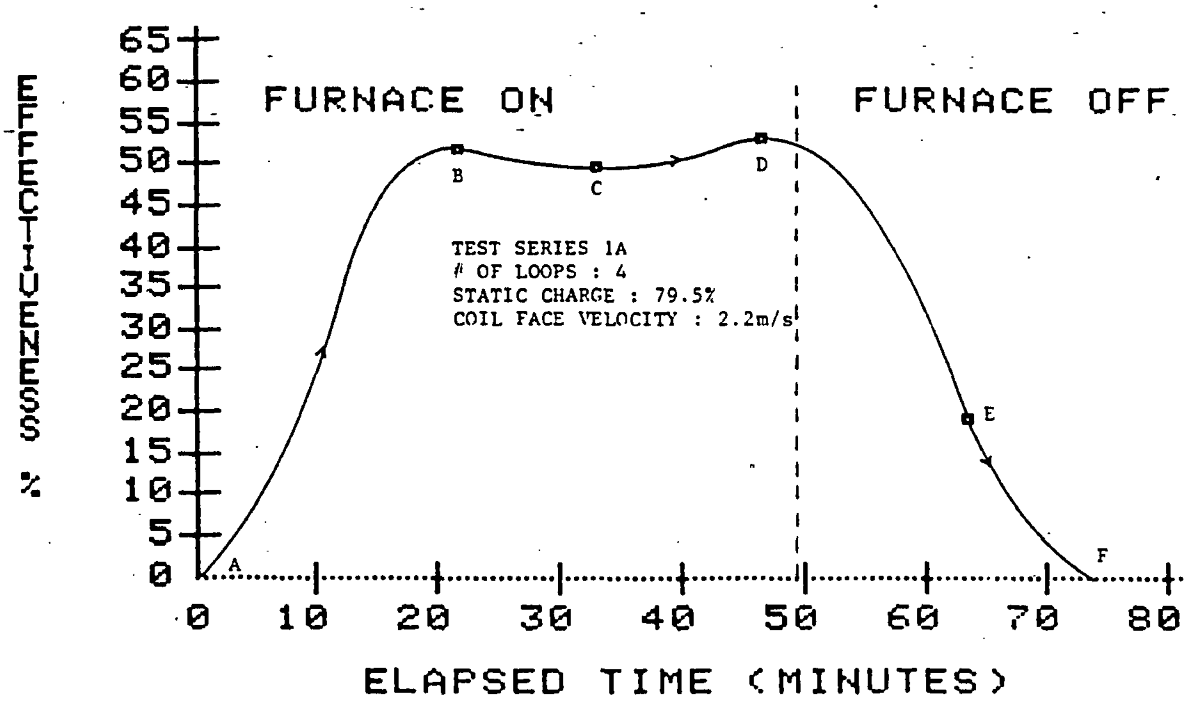

FIGURE 7.2.1
TRANSIENT THERMAL RESPONSE OF THE TSHE APPARATUS

determined that quasi- steady state conditions were achieved inside of 20 minutes provided no other system parameters were changed. As a result of this test, it was decided that a one hour time interval between data collections throughout the subsequent experiments would provide ample time for steady state to be fully established.

7.3 DATA ANALYSIS

The effectiveness of the system, which is the measure of performance used in this study, was calculated from the difference in average energy flowrates at each of the 4 measurement stations taking into consideration the effect of humidity, static pressure, and heat loss through the ducting walls.

The equations below summarize the method used to calculate the effectiveness of the system in the data acquisition program.

$$EFFECTIVENESS = \frac{Qevap + Qcond}{2} * \frac{1}{Qmax} * 100$$

where:

$$Qevap = \frac{Mue + Mde}{2} (hue - hde)$$

$$Qcond = \frac{Mdc + Muc}{2} (hdc - huc)$$

$$Qmax = \frac{Mue + Muc}{2} (hue - huc)$$

ue - Upstream Evaporator
de - Downstream Evaporator
uc - Upstream Condenser
dc - Downstream Condenser

M - Average mass flow rate
Q - Average energy flow rate

and:

$$h_{ij} = \left[\frac{\sum_{i=1}^{18} m_i h_i}{\sum_{i=1}^{18} m_i} \right]_{ij} \qquad \text{(average enthalpy of air at location ij)}$$

$$M_{ij} = \left[\sum_{i=1}^{18} m_i \right]_{ij} \qquad \text{(total mass flow rate of air at location ij)}$$

m_i - mass flow rate of air through 1/18 area rectangle 'i'

h_i - enthalpy of air through 1/18 area rectangle 'i'

A detailed review of the equations and methodology used for the analysis of the experimental data is available in Appendix F. The computer data acquisition program flowchart , program listing, and table of variable names can be found in Appendix G.

7.4 EXPERIMENTAL UNCERTAINTY

The Kline and McClintock {14} method of uncertainty analysis was applied to this study and resulted in an uncertainty in the effectiveness of 15% . As reported in most experimental studies in the literature, inaccuracies in the measured data of 10 to 30% are not uncommon using the technology presently available. Appendix H contains a brief description of the Kline and McClintock method used and summarizes in tabular form, the different parameters and their estimated and calculated uncertainty.

8. RESULTS AND DISCUSSION

8.1 FOUR LOOP SYSTEM

8.1.1 2.2 M/S COIL FACE VELOCITY

In order to study the hysteresis performance behaviour of the 4 loop (2 row of tubes per loop) TSHE system, test sequence 1 through 8 (Table 7.2.1) were carried out in which the overall temperature difference of the TSHE apparatus was first increased incrementally, then decreased for 3 different heat exchanger coil face velocities. Although the test apparatus was not designed to specifically study the incipient point of nucleate boiling and thus hysteresis, a good indication of the magnitude of superheat required for an industrial size system and how the hysteresis phenomena can cripple a TSHE system is easily seen from the performance data collected in this study.

The effect of the wall superheat required to initiate boiling on the performance of the TSHE system is seen in the Effectiveness versus Overall Temperature Difference plot of Figure 8.1.1.1 . It is clear from this plot that nucleate boiling, with its characteristically high heat transfer coefficient, was not initiated until a hot-to-cold duct air stream temperature difference of approximately 13 Celsius degrees was achieved. This amount of superheat is comparable with other studies {8} in which a superheat temperature of 13.8 to 16.7 Celsius degrees was found to be necessary to start the boiling process.

As the TSHE system was further heated past the incipient boiling point, the performance, or effectiveness, is seen to improve rapidly and soon level out to become almost constant

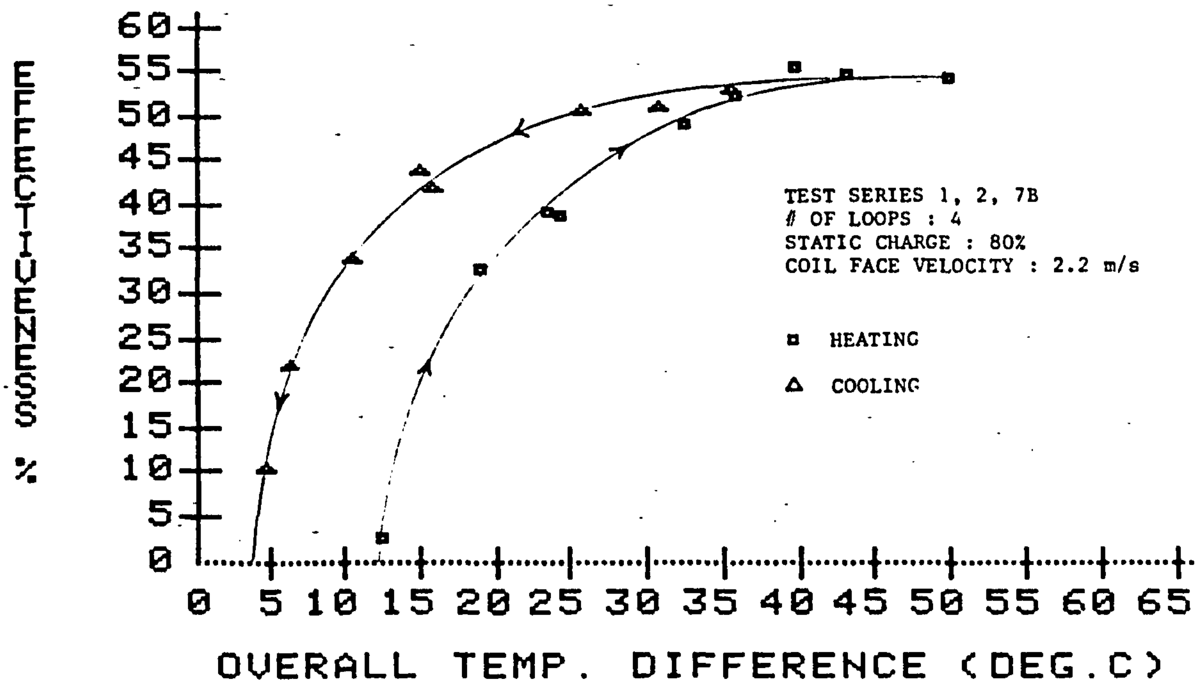

FIGURE 8.1.1.1
EXPERIMENTALLY DETERMINED 4 LOOP HYSTERESIS
ENVELOPE WITH A 2.2 M/S COIL-FACE VELOCITY

above a 40 Celsius degree overall temperature difference. Upon decreasing the overall temperature difference, a new performance curve was generated which clearly exceeded that of the heating case. These two curves form a hysteresis envelope within which the system must operate.

The position along a vertical, liquid filled, constant heat flux, evaporator tube where nucleate boiling will first take place is at or near the tubes static liquid level height. Upon subjecting the tube to an increased heat flux, the elevation where incipient boiling occurs will decrease and gradually move down the tube to a new equilibrium position. If the heat flux is increased substantially enough, the boiling point will move to the bottom of the tube and thus, the majority of the possible nucleation sites along the tubes inside wall surface will have been developed. Under this condition, an increase in the heat flux supplied will yield a negligible increase in the amount of heat energy transported to the working fluid. A limit will be reached when dryout starts to occur in the evaporator tubes.

When the nucleation sites along the tube inner wall surface have been formed, they will continue to be active for wall temperatures below that necessary to create them. For this reason the cooling performance curve of Figure 8.1.1.1 is seen to exceed the heating curve but retains the same shape since the boiling point within the evaporator tubes is now rising during cooling. A bubble formed inside a nucleation site will be quenched when the tube wall temperature is reduced to a point where the corresponding saturation pressure within the bubble no longer exceeds the combined effects of the local liquid pressure and the interfacial surface tension force between the bubble and fluid per unit of bubble cross sectional area. When the overall temperature difference decreases below approximately 4 Celsius degrees it was found that most of the nucleation sites were extinguished.

To more fully understand the behavior of the system in the region of the hysteresis envelope, two test sequences were conducted in which the overall temperature difference was cycled between fixed temperature limits.

The first of these tests, Test sequence 2A, was aimed at examining the performance of the system in the region where the onset of boiling took place. For this reason, the overall temperature difference was first increased stepwise from 0 to 30 Celsius degrees , then decreased from 30 to 5 Celsius degrees , and finally increased again to 13 Celsius degrees . This cycling of the overall temperature difference produced the performance curve shown in Figure 8.1.1.2 and clearly illustrates that upon reheating of the system from 6 to 13 Celsius degrees {points `G` to `H`}, the performance of the TSHE had been retarded. This reduction in the effectiveness of the system at point `H` is thought to be caused by the previous quenching of nucleation sites.

The second experiment, Test sequence 8, was repeatedly cycled between overall temperature differences of 23 and 35 Celsius degrees. Figure 8.1.1.3 shows the results of this test which indicate that unlike test sequence 2A, the performance was not retarded upon reheating but instead remained on a performance curve which was dictated by the maximum value of the overall temperature difference last achieved in the system.

It is postulated that when the overall temperature difference is reduced by more than 13 Celsius degrees C as was done in Test sequence 2A, some nucleation sites are quenched and hence a reduction in the performance of the TSHE occurs (In Test sequence 8, the overall temperature difference was cycled between 23 and 35 Celsius degrees which is a reduction of only 12 Celsius degrees. It is suspected that only a very small number if

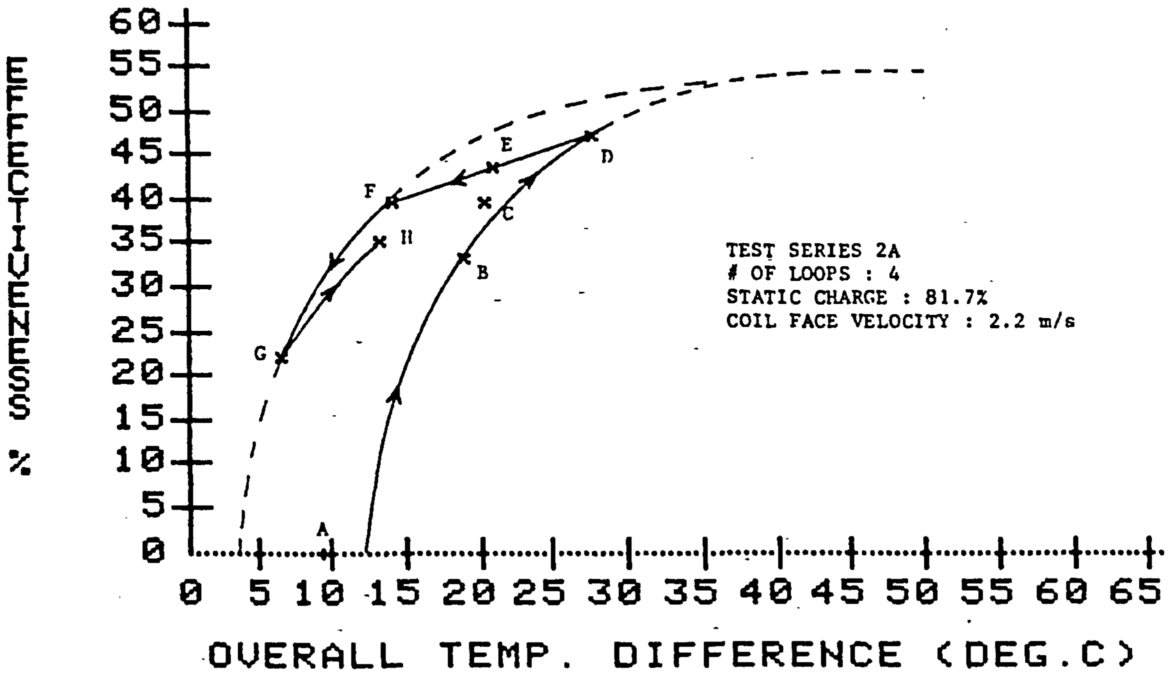

FIGURE 8.1.1.2
CYCLED OVERALL TEMPERATURE DIFFERENCE
BETWEEN 6 AND 30 CELSIUS DEGREES

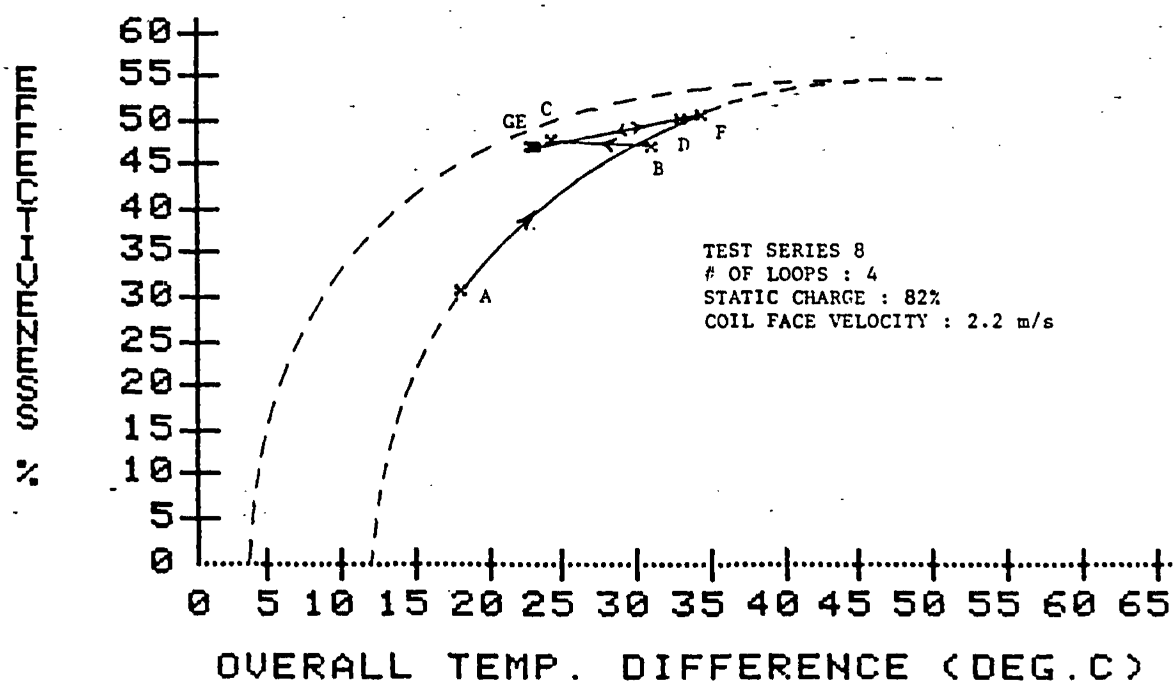

FIGURE 8.1.1.3
CYCLED OVERALL TEMPERATURE DIFFERENCE
BETWEEN 23 AND 35 CELSIUS DEGREES

any, of nucleation sites were destroyed in this test and thus, performance was not appreciably affected.)

If the performance line connecting these two points in test sequence 8 (F and G) were extended to the cooling curve, a temperature difference of approximately 15 Celsius degrees would be encountered and it is suspected that the quenching of nucleation sites would take place.

Upon a closer examination of Test sequence 2A, it was observed that the overall temperature difference between points `D` and `F` was 13.5 Celsius degrees which locates point `F` extremely close to the original cooling curve of Figure 8.1.1.1 . Since the TSHE system was cooled below point `F` to point `G` in test sequence 2A, an overall temperature difference reduction of 20.85 Celsius degrees took place and thus, a significant number of nucleation sites were most likely quenched.

8.1.2 1.3 AND 3.1 M/S COIL FACE VELOCITY

In order to determine the effect on the hysteresis envelope of increasing the heat exchanger coil face velocities, two further test sequences were conducted in which a coil face velocity of 1.3 and 3.1 m/s were imposed .

The results of these tests are plotted in Figures 8.1.2.1 and 8.1.2.2 and clearly show that the hysteresis envelope is preserved for higher and lower air velocities. Although not planned, the discrepancy between these figures and that of Figure 8.1.1.1 was caused by beginning the experiments before the TSHE system had completely cooled to room temperature and quenched all of the previously created nucleation sites. For this reason,

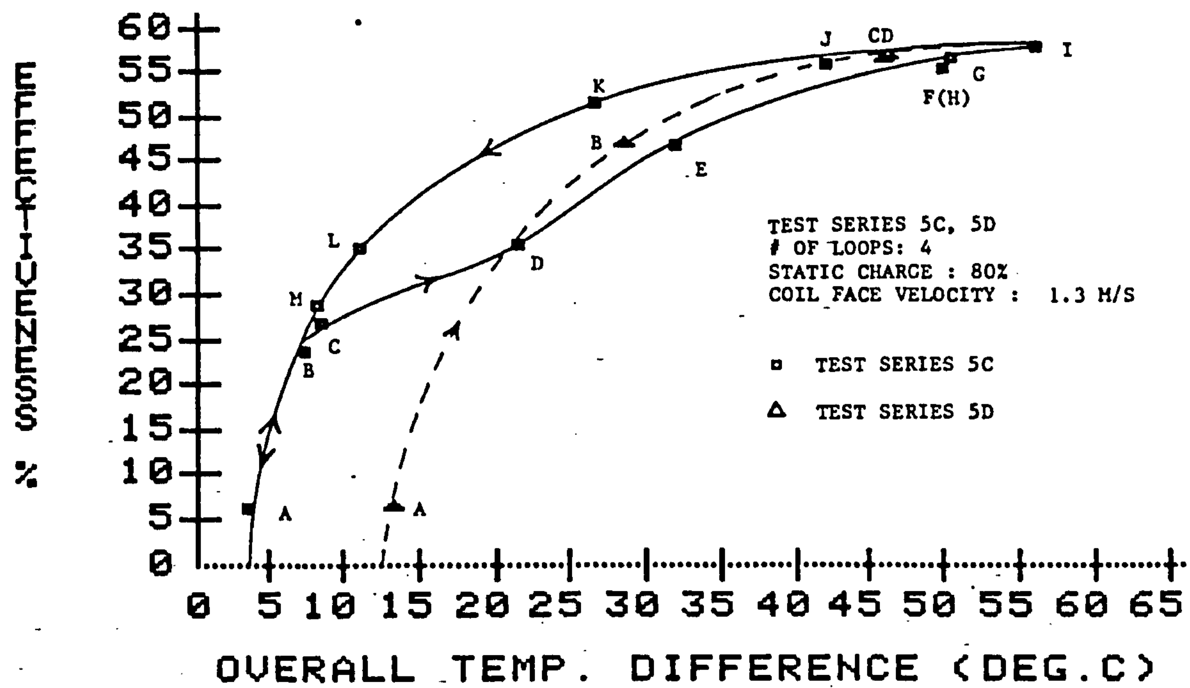

FIGURE 8.1.2.1
EXPERIMENTALLY DETERMINED 4 LOOP HYSTERESIS ENVELOPE WITH 1.3 M/S COIL FACE VELOCITY

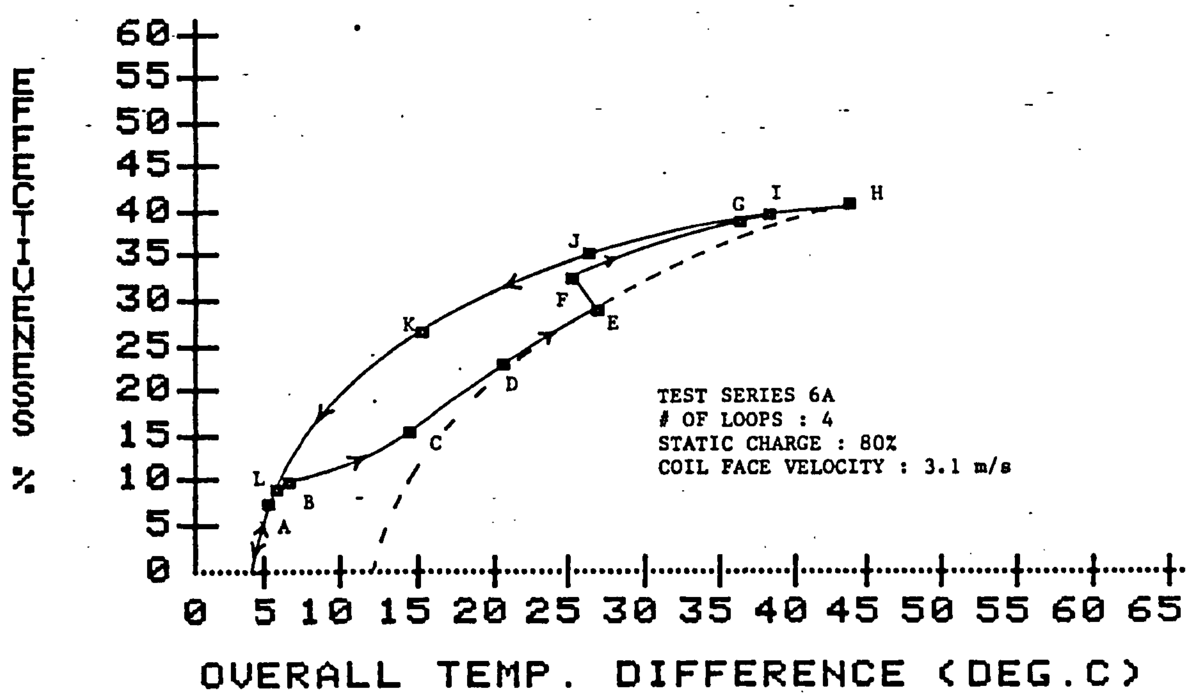

FIGURE 8.1.2.2
EXPERIMENTALLY DETERMINED 4 LOOP HYSTERESIS ENVELOPE WITH 3.1 M/S COIL FACE VELOCITY

some of the sites were still present and active in the evaporator tubes and were transporting large amounts of energy when the system was slightly heated. To ensure that a complete hysteresis envelope was still present, test sequence 5D shown on Figure 8.1.2.1 was carried out in which the system was sufficiently cooled prior to testing to quench any residual nucleation sites. This curve shows that the shape of the heating and cooling curves shown in Figure 8.1.1.1 for a face velocity of 2.2 m/s is maintained with a face velocity of 1.3 m/s.

From Figures 8.1.1.1 through 8.1.2.2, it is clear that the performance of the TSHE system can be severly affected by the incipient boiling behaviour of a refrigerant. For the system tested, it would appear that operation above an overall temperature difference of 40 Celsius degrees would yield the most consistent and reliable performance. Operation with temperature differences between 40 and 13 Celsius degrees would be satisfactory only if the system was periodically subjected to a higher temperature difference so as to ensure closer operation to the cooling curve. The system should not be operated with temperature differences below 13 Celsius degrees since the performance in this region is extremely sensitive to overall temperature difference fluctuations and may in fact never start.

8.2 SINGLE LOOP SYSTEM

In order to study the performance of a single loop, 2 row of tubes per loop TSHE system, refrigerant was drained from 3 of the loops in the 4 loop test facility apparatus. To determine which of the 4 loops should contain the refrigerant for testing of a single loop, a decreasing flux test was conducted for loops 1, 2, and 4. Loop 3 was not tested since it was determined earlier that this loop had developed a leak which could not be located. Figure 8.2.1 shows the results of these tests for all three loops and indicates that there is good agreement in the performance between the different loops. For this reason, Loop 1 was arbitrarily chosen for use in further studies.

To determine the influence of the hysteresis envelope on the performance of a single loop system, Figure 8.2.2 shows a plot of effectiveness versus the overall temperature difference for Test sequence 13A in which the coil face velocity was 2.2 m/s. A hysteresis envelope is again seen to significantly influence the system performance in the overall temperature difference region of 5 to 40 Celsius degrees. Since the performance curves of the 4 loop system found earlier suggest that nucleate boiling was initiated near an overall temperature difference of 13 Celsius degrees , it is believed that point 'A' of this test was erroneous and so was neglected in the subsequent analysis. The heating curve of test sequence 13 shown in Figure 8.2.3 was conducted under the same conditions and supports this decision.

As in the 4 loop tests explained earlier, once boiling had been initiated, the effectiveness of the system again rose quickly and leveled out near an overall temperature difference

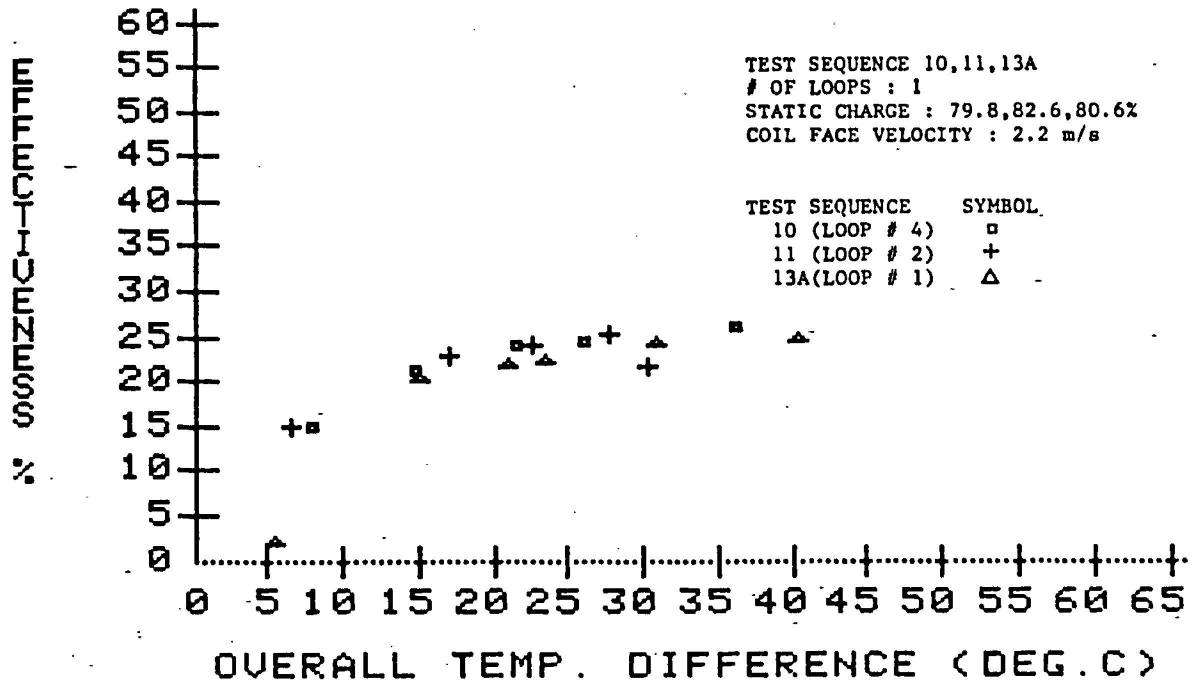

FIGURE 8.2.1
PERFORMANCE COMPARISON BETWEEN LOOPS 1, 2, AND 4
(COOLING CURVES)

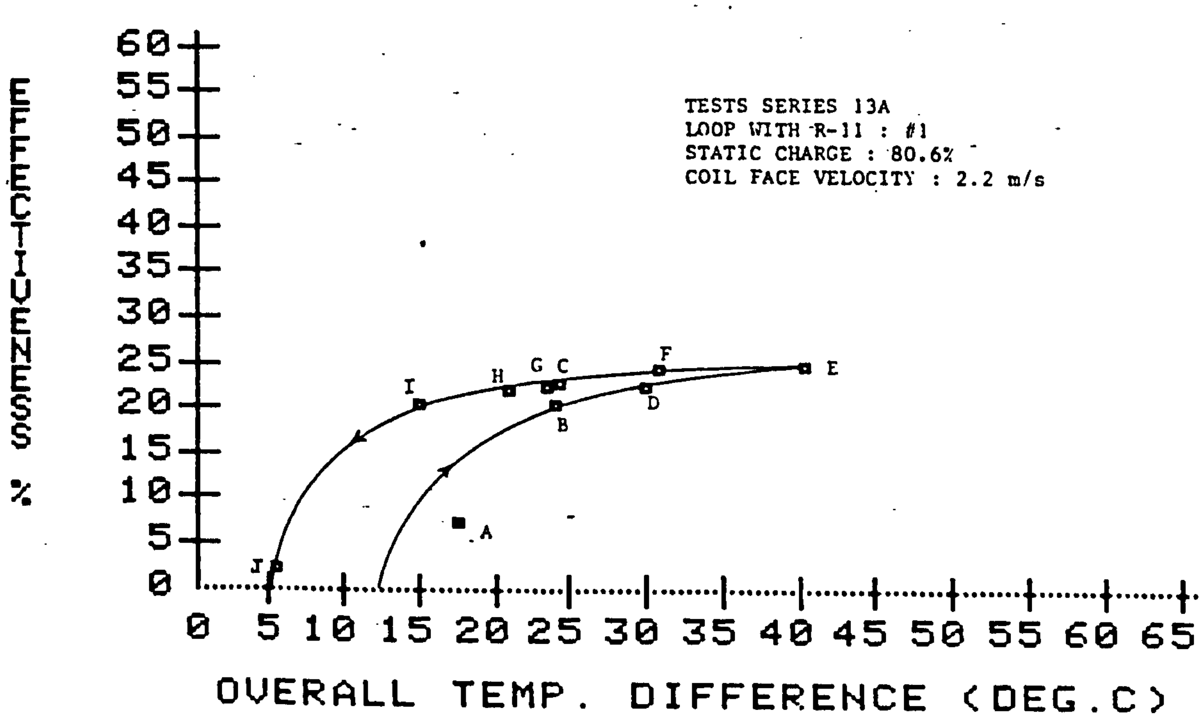

FIGURE 8.2.2
EXPERIMENTALLY DETERMINED SINGLE LOOP HYSTERESIS
ENVELOPE WITH 2.2 M/S COIL FACE VELOCITY

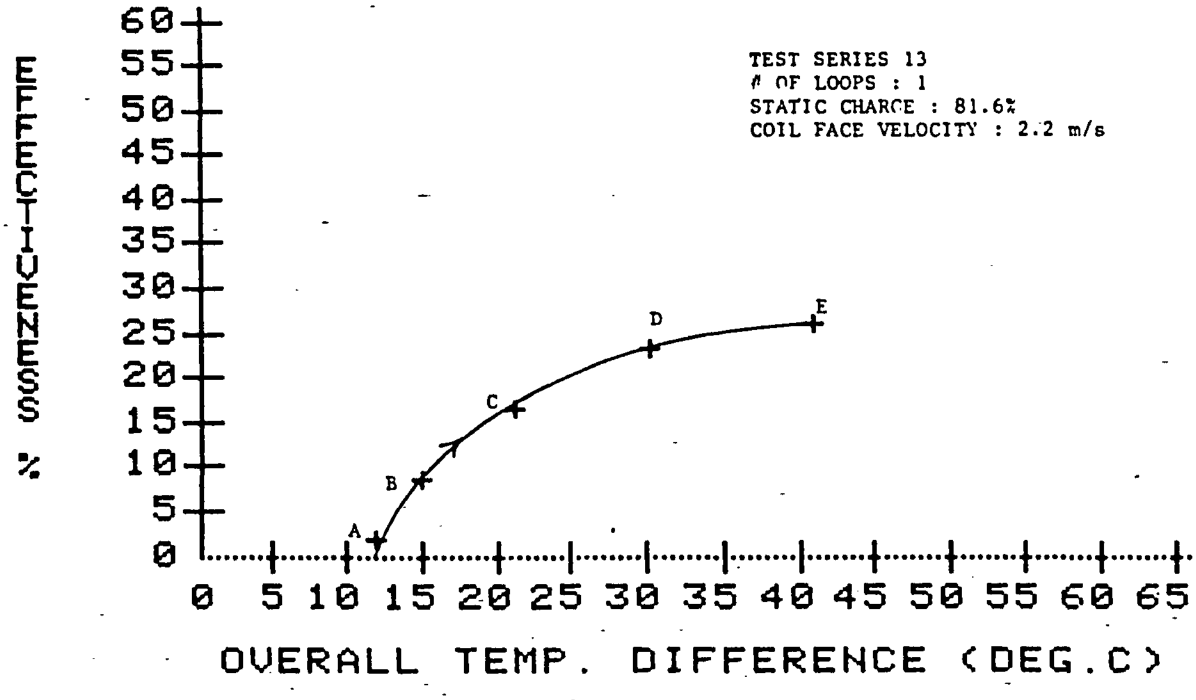

FIGURE 8.2.3
EXPERIMENTALLY DETERMINED SINGLE LOOP HEATING CURVE WITH 2.2 M/S COIL FACE VELOCITY

of 40 Celsius degrees. Using the composite results of test sequences 10 through 13A, it was found that the performance of a single loop TSHE could be modelled with moderate accuracy by the equation:

$$Eff(1) = Emax\{1 - e^{(OTDIB - OTD)/C}\} \qquad \text{Eqn. 8.2.1}$$

where:

- Emax – maximum effectiveness for each air flow rate (%)
- Eff(1) – effectiveness of the single loop TSHE (%)
- OTDIB – overall temperature difference at incipient boiling {degrees C}
- OTD – overall temperature difference {degrees C}
- C – heating/cooling constant

Typical values determined from Test 13A for Emax, OTDIB, and C are listed in table 8.2.1 below.

TABLE 8.2.1
TYPICAL VALUES USED IN EQUATION 8.2.1

	HEATING	COOLING
Emax	25.5 *	25.5 *
OTDIB	12.0	4.0
C	7.0	6.0

* - Values are for the 2.2 m/s heat exchanger coil face velocity.

Figure 8.2.4 shows a comparison between the experimentally determined performance and the performance calculated using equation 8.2.1 .

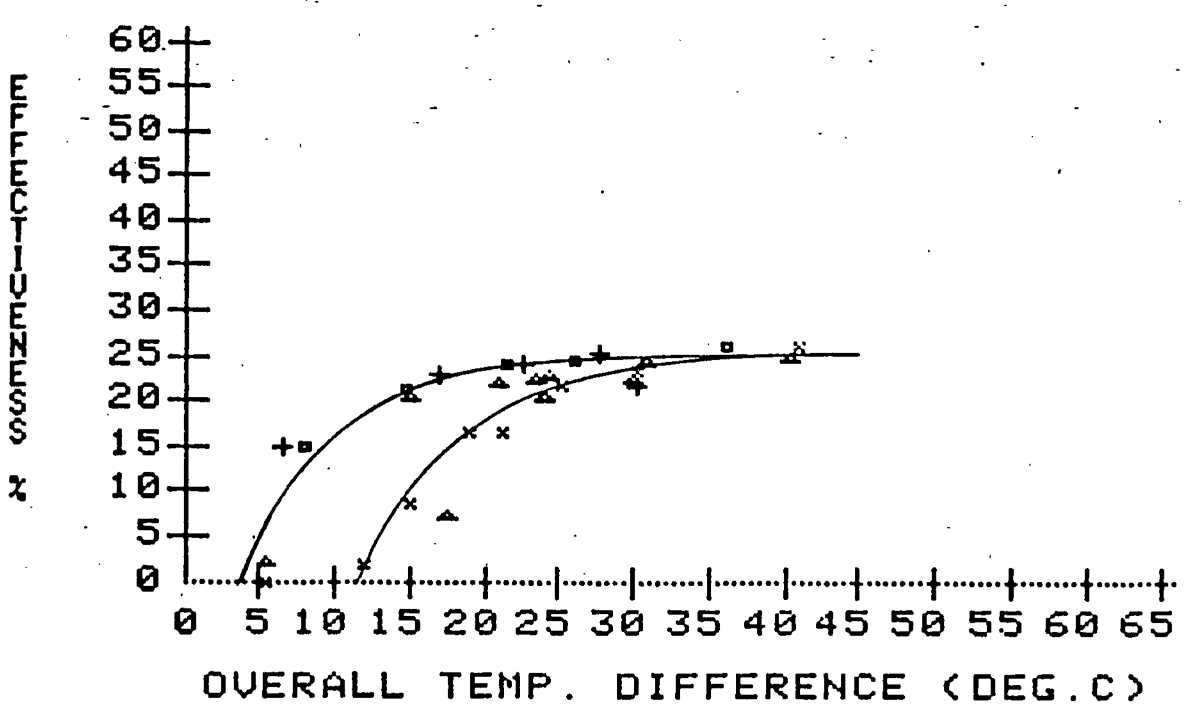

FIGURE 8.2.4
EXPERIMENTALLY DETERMINED SINGLE LOOP DATA
AND THE CURVES GENERATED USING EQUATION 8.2.1

8.3 FOUR LOOP PREDICTION

Using the data obtained from the single loop test sequence 13A, an Apple basic program was written to iteratively calculate the performance of 2, 3, and 4 loop system configurations taking into account the actual effectiveness of the loop for the overall temperature difference imposed. The single loop data was least squares error curve fitted to a polynomial and used in the iteration to produce the plot of Figure 8.3.1 . The software written and the methodology used in this analysis can be found in Appendix I.

Reasonably good agreement was found when the computer predicted performance of a 4 loop system configuration was compared to the earlier determined, experimental 4 loop test

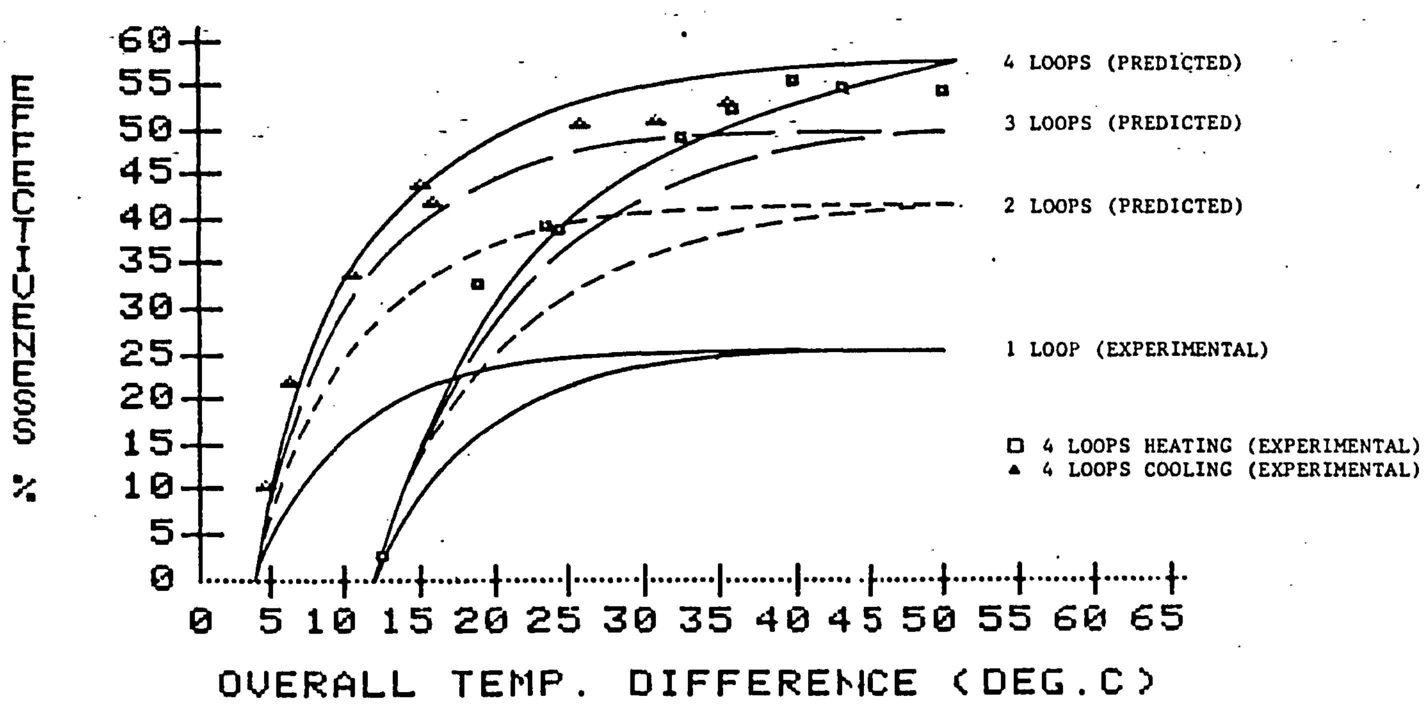

FIGURE 8.3.1
PREDICTED 2, 3, AND 4 LOOP PERFORMANCE

results. Figure 8.3.1 shows this comparison and validates the prediction program's ability to give a quick indication of the magnitude of the performance expected for multiloop TSHE systems. A program of this type may be useful when estimating the cost of new and retrofit TSHE systems.

8.4 COMPARISON WITH EARLIER WORK

A variety of experiments were previously carried out by Kosnik and Bertoni on the same TSHE test apparatus used in this study . Although the majority of the instrumentation was upgraded between the two studies, a good comparison between the results obtained here and those of Kosnik and Bertoni was achieved. Figure 8.4.1 is a composite plot of Effectiveness versus overall temperature difference for the 3 different coil face velocities

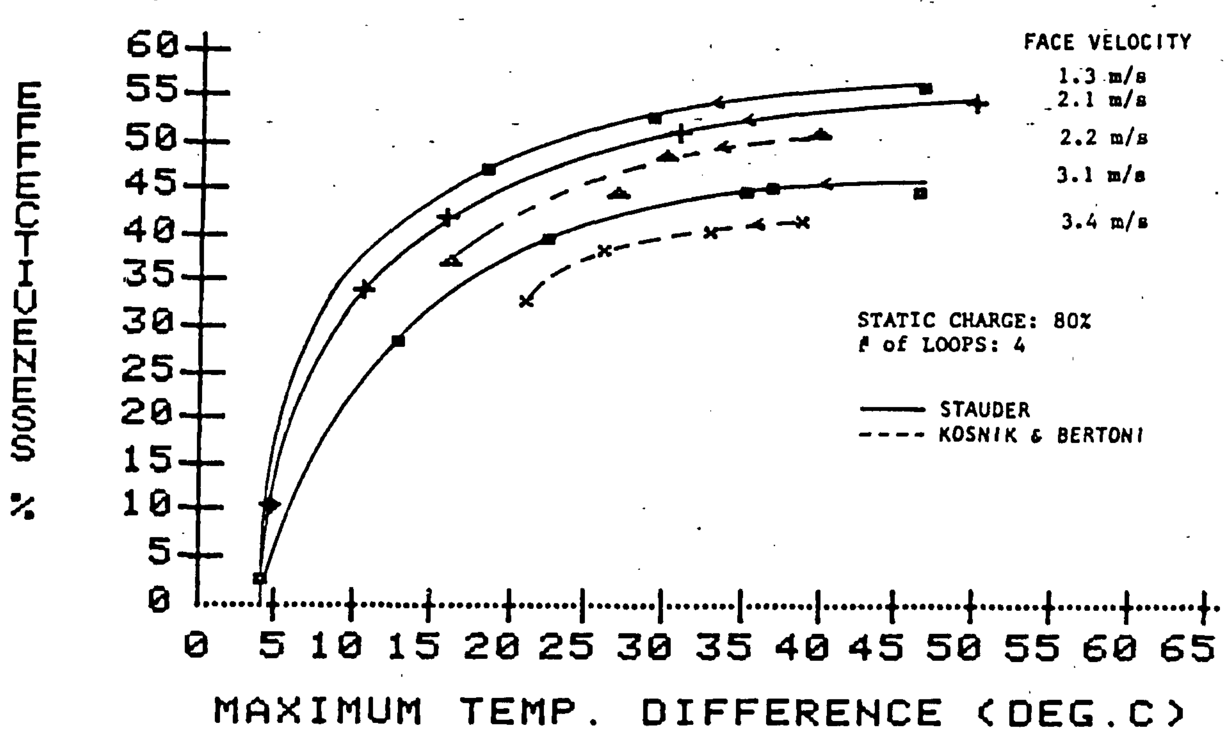

FIGURE 8.4.1
COMPARISON OF RESULTS WITH EARLIER WORK OF KOSNIK AND BERTONI

used in this study and the two used by Kosnik and Bertoni. It is easily seen that the general form of the curve is essentially the same with the difference being a vertical shift in the position of the curves. This shift may be due to the difference between the velocity profiles used in each experiment since the degree of turbulence in the ducting was enhanced in this study to give a more uniform velocity profile.

8.5 EXPERIMENTAL PROBLEMS AND DELAYS

Some tests needed to be repeated because non-condensible gases were found to be present in one or more of the loops. Non-condensible gases in steam-boiler systems are known to reduce performance substantially. ASHRAE {15} reported a reduction in the heat transfer coefficient of 70% when 2.89% air by volume was introduced into the system.

Before testing began, the TSHE system apparatus was carefully inspected for refrigerant R-11 leaks, and repairs were immediately made on those found. Unfortunately, it was not possible to locate all of the leaks before testing commenced.

To reduce the possibility of having air drawn into the refrigerant flow so that tests could be conducted, the evaporator vapour temperature was continuously kept above 25 degrees C between tests since the vapour pressure of R-11 at this temperature is greater than atmospheric pressure and as such, the system would lose refrigerant instead of gaining a non-condensible gas. Although this method worked fairly well, it was suspected that non-condensible gases would sometimes be drawn into the system at the end of a test when small overall temperature differences were required. It is believed now, that a leak was present somewhere within the condenser heat exchanger since this was the location of lowest pressure in the system and one of the areas where inspection for leaks was difficult.

It is strongly recommended that the test apparatus be thoroughly inspected for refrigerant leaks and repairs made. This may involve the extraction of one , or both of the heat exchanger coils in order to properly effect repairs.

When charging the system with refrigerant R-11, either from the reservoir or from an external fresh tank, it was often difficult to obtain an equal level of liquid refrigerant resident within each of the 4 loops. Although the liquid side of the loops could be interconnected, the vapour side could not and as such, the liquid level would be determined by the total pressure present in the loop thus indicating the presence of non-condensible gases. Once repairs have been made to the refrigerant leaks throughout the system, the leveling of the static charge will not be as difficult since the vapour pressure in each loop will be closer to that of the pure refrigerant. To help level the static charge in the system when very small amounts of non-condensible gases are present, installation of 4 interconnecting valves located off the evaporator vapour header is suggested. This would allow the pressures in each of the 4 loops to be identical when the valves are opened and thus, enable the liquid levels between loops to quickly equalize.

9. CONCLUSIONS

The following conclusions about the performance behavior of thermosiphon heat exchangers can be drawn from this study:

1. The reported high degree of wall superheat required to initiate boiling can severely affect the performance of the TSHE for small overall temperature differences . For larger temperature differences, the effectiveness of the system was found to be essentially constant for all coil face velocities studied. With the 4 loop and single loop, 2 row of tubes per loop system configuration investigated here, the onset of boiling was seen to occur near an overall temperature difference of 13 Celsius degrees and the effectiveness was observed to be essentially constant above an overall temperature difference of 40 Celsius degrees for a coil face velocity of 2.2 m/s .

2. The quenching of nucleation sites brought about by a decreasing heat flux {cooling} was seen to retard the performance of TSHE system when reheating for small overall temperature differences. The effect of the quenching of sites was clearly seen in Figure 8.1.1.2 where the overall temperature difference was reduced from 27 Celsius degrees to 7 Celsius degrees.

3. It was postulated that the quenching of nucleation sites will occur whenever the overall temperature difference of the system is reduced by the same amount required to originally initiate the boiling process. Figure 9.1 shows how this postulate can affect the performance of the TSHE system in the region of the envelope above the incipient boiling overall temperature difference. When the TSHE system is initially heated

from an overall temperature difference of 0 Celsius degrees to point A{26 C} and later to point B{39 C}, the performance will follow the characteristic increasing flux {heating} curve as was shown in Figure 8.1.1.1 . Upon cooling of the system from point B{39 C} to C{26 C} , the performance will remain essentially constant and close to the maximum possible performance. Further cooling from point C{26 C} to D{13 C} will yield the same performance characteristics of a decreasing flux {cooling} curve with performance being reduced by 27%. Upon reheating from point D{13 C} to A{26 C}, the performance is seen to migrate in a straight line to the original increasing flux {heating} curve and improve only slightly {5%}.

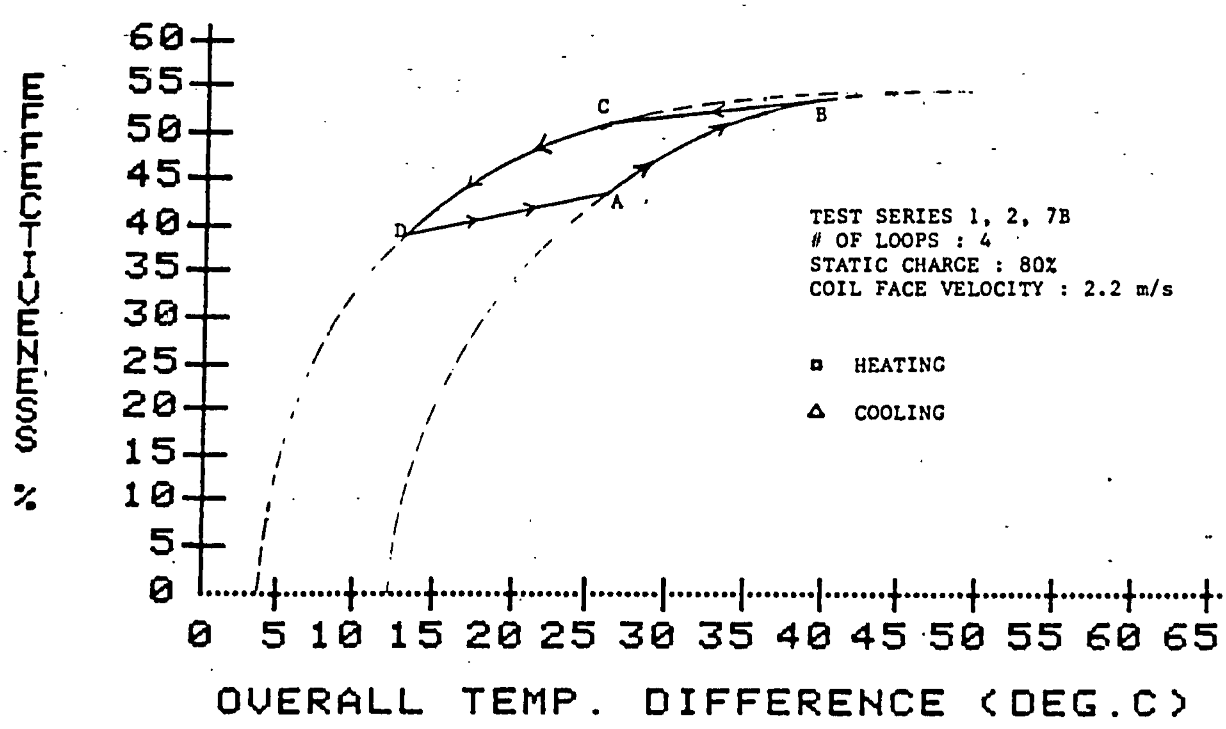

FIGURE 9.1
POSTULATED PERFORMANCE OF 4 LOOP THERMOSIPHON SYSTEM

From this postulated cycle, two conclusions are readily drawn.

1. A minimum effectiveness of 40% {73% of the maximum effectiveness} will be maintained when the TSHE system is first heated to an overall temperature difference of at least two times the amount necessary to initiate boiling when operating in the region of the envelope above the incipient boiling point overall temperature difference. For the TSHE system studied here, a minimum overall temperature difference of 26 Celsius degrees would be necessary.

2. A negligible decrease in performance may be expected when operating with a minimum overall temperature difference of two times the incipient boiling point temperature provided that the system was first heated to an overall temperature difference of at least 3 times the incipient boiling point temperature. In the 4 loop TSHE system studied here, a minimum overall temperature difference of 39 Celsius degrees would first be required to ensure that near maximum performance would result when operating in the region above a 26 Celsius degree overall temperature difference.

It is thus suggested from these conclusions that operation of the TSHE system should be restricted to overall temperature differences greater than two times the temperature difference required to initiate boiling. In this way, the performance of the TSHE system will not be drastically affected by small overall temperature difference fluctuations. If possible, operation above three times the overall temperature difference required for the onset of boiling is recommended since the performance is optimized in that region.

If operation is required with overall temperature differences of below two times the incipient boiling point temperature difference, care should be exercised to strictly avoid the region where the overall temperature difference is less than that required to initiate boiling since the performance of the TSHE in that region was found to be very dependent on past thermal history.

4. The single loop performance was modelled with good accuracy by an equation of the form:

$$\text{Eff} = \text{Emax}(1 - e^{(\text{OTDIB} - \text{OTD})/C}) \qquad \text{eqn. 9.1}$$

where Emax, OTDIB, and C are constants depending on the air flow rate, and past thermal history.

5. Using the single loop experimental data, an iterative computer routine was developed which could predict the performance of 2, 3, and 4 loop systems with good accuracy.

6. The effectiveness of a multiloop system , when operating with a given overall temperature difference, will increase with the addition of loops to the system. Figure 9.2, a plot of effectiveness versus the number of loops for a 40 C overall temperature difference, clearly shows this trend.

A careful examination of the computer generated data for an overall temperature difference of 40 Celsius degrees showed that the effectiveness of the individual loops in the system were equal. As more loops were added to the system, the temperature difference across each loop approached the minimum temperature difference necessary for boiling. The maximum effectiveness for an infinite number of loops was determined to be;

$$\text{Eff(max)} = \frac{\text{TDo} - \text{TD1}}{\text{TDo}} \qquad \text{eqn. 9.2}$$

Where : - TDo is the overall temperature difference of the system

- TD1 is the minimum temperature difference across one loop in the system required for boiling.

Figure 9.3 is a plot of equation 9.2 and shows the maximum effectiveness that may be achieved using an infinite number of loops at a particular system overall temperature difference. Note that there were two curves generated, one for heating (TD1=12 C) and one for cooling (TD1=4 C) since each process requires a different overall temperature difference to initiate or maintain boiling.

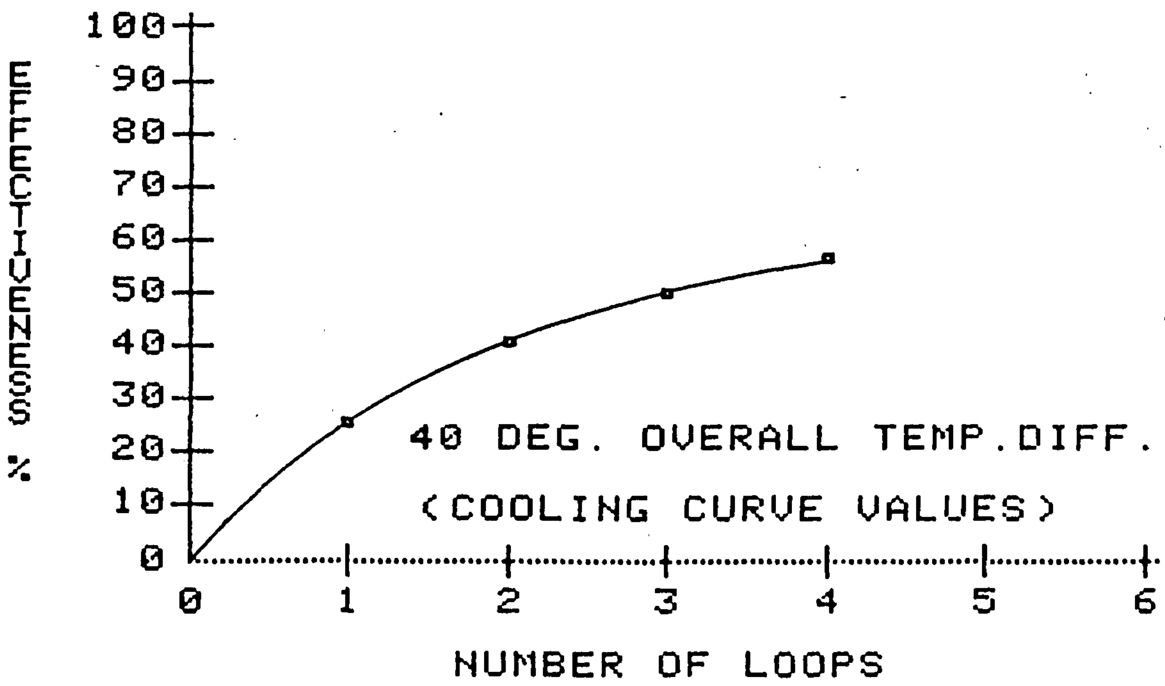

FIGURE 9.2
COMPARISON OF EFFECTIVENESS BETWEEN MULTILOOP SYSTEMS

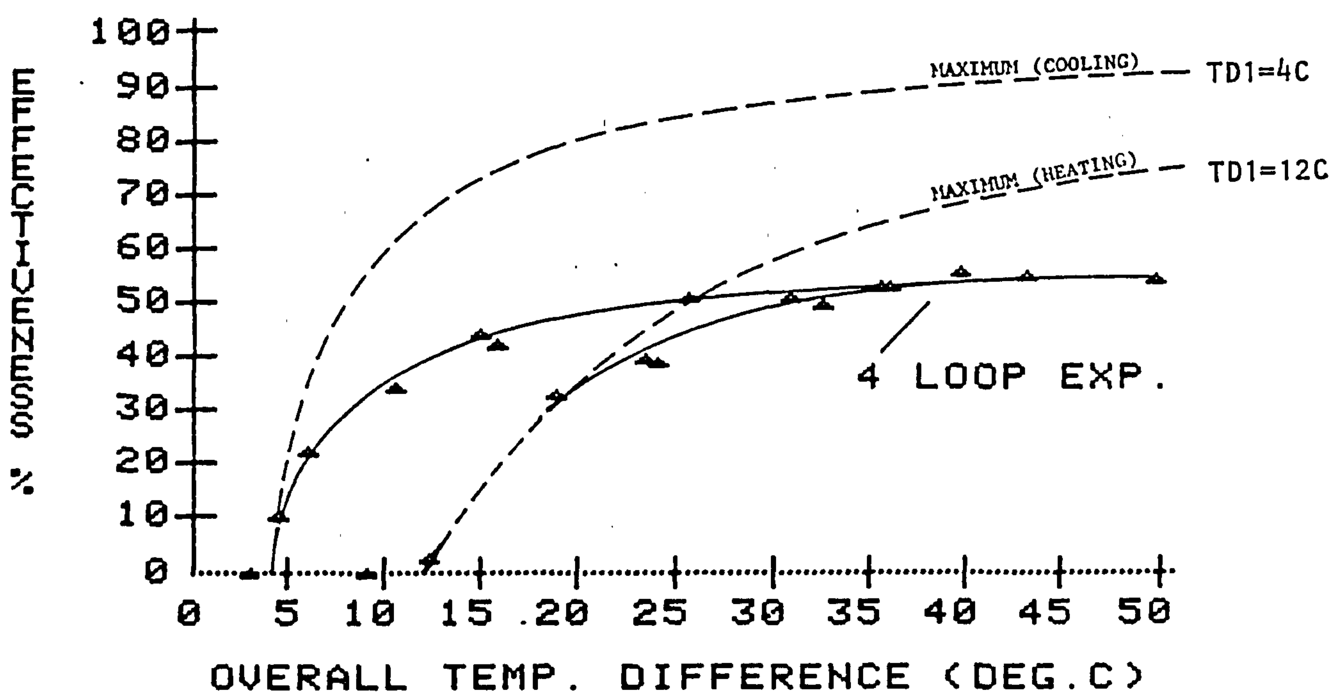

FIGURE 9.3
COMPARISON OF MAXIMUM POSSIBLE EFFECTIVENESS FOR AN INFINITE NUMBER OF LOOPS WITH EXPERIMENTAL RESULTS FOR 4 LOOPS

In summary, it would appear that the thermosiphon coil loop run-around heat exchanger system is a viable heat recovery device for industrial applications where the operating temperatures clearly exceed the region where the effects of hysteresis are predominate. It is suggested that further investigations into multi-fluid, multi-loop systems could improve the feasibility and attractiveness of these devices for a wider variety of applications.

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APPENDIX A

DUCT VELOCITY AND PRESSURE MEASUREMENTS AND DAMPER CONFIGURATIONS

DUCT AIR VELOCITY PROFILE

In order to obtain a more accurate indication of the air flow rate and later the mass flow rates of air in the ducting, the duct inside area was divided into 18 imaginary, equal area rectangles as shown in Figure A.1 . Velocity readings were taken at the center of each of the 18 equal area rectangles with the calibrated Velometer and probe to achieve a 2 dimensional profile of the air's velocity distribution. In total,4 velocity profiles were measured within the ducting, one at each of the measurement stations located before and after both the condenser and evaporator heat exchangers. A procedure consisting of three steps was taken to obtain the velocity profiles with the Velometer and probe :

1. The profile was started at the rectangular area which permitted the least amount of the probe to be in the duct.
2. The clear plastic Velometer hoses were shaken twice to help stabilize a proper velometer reading.
3. The profiles were read in the order of ; downstream evaporator, upstream evaporator, downstream condenser, upstream condenser.

The velocity data obtained from this procedure is summarized in Table A.1 .

Since the velocity profiles were obtained while the system had no overall temperature difference across the heat exchangers, the volume flow rate of air through the system was considered constant. For this condition, the average velocity for each measurement station should be the same since the areas of the ducting and the density of the air were constant. Unfortunately, it was found that the average velocities differed slightly between stations. It was thought that the Velometer readings obtained along the duct wall did not accurately estimate the average velocity present in that 1/18 area of duct. To correct for this, a detailed one inch increment velocity profile was taken along a traverse through the center of the duct for each of

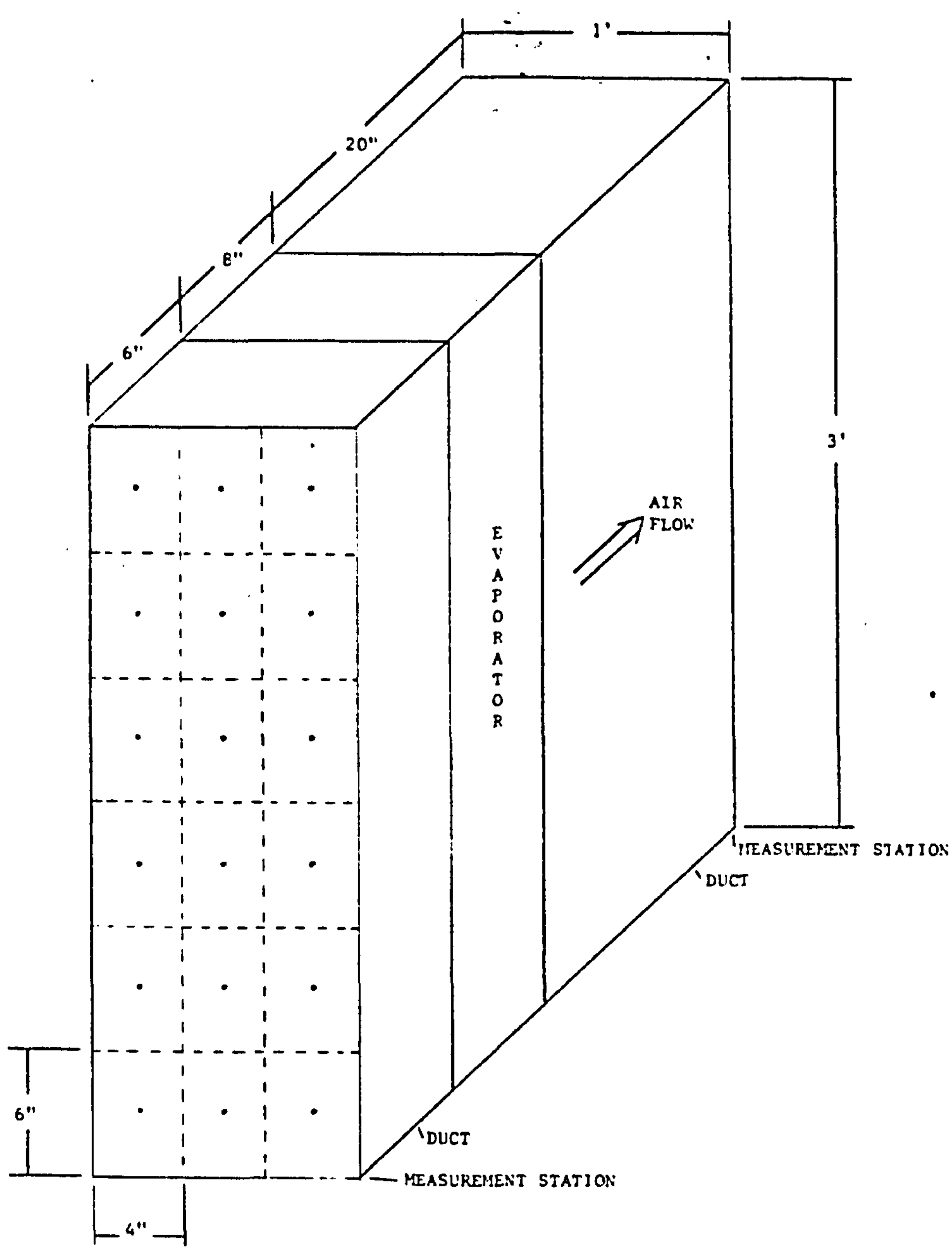


FIGURE A.1
GRID DIVISION USED TO OBTAIN VELOCITY DISTRIBUTIONS

the 4 measurement stations. From these detailed velocity profiles which are shown in Figures A.2 through 5, a ratio of the detailed, 1 inch increment average velocity for 1/18 area of duct divided by the measured velocity for that same area {taken at the center} was determined and was later used to derive a corrected velocity profile. Although this method assumes that the velocity profile was only 1 dimensional, it was believed, through spot checks with the Velometer and probe, that a more accurate representation of the velocity distribution was obtained since the average velocities were now in more agreement with each other.

From these corrected velocity profiles, the normalized profiles were calculated by first converting the corrected Velometer readings to the actual velocity by means of the Velometer calibration curvefit, then dividing all 18 actual velocities by one of 4 reference velocity readings obtained from a point in the center of each of the respective measurement stations. Traverse position numbers 7, 27, 50 and 67 found on the apparatus were consistently used as reference locations. The normalized profile was then incorporated into the computer data acquisition program to calculate the velocity of the air for any 1/18 area of duct given its measurement station and reference velocity reading. Both the corrected velocity profiles and the normalized profiles can be found at the end of this Appendix.

To further correct the velocity profile so that all 4 measurement stations gave the same average velocity, a correction factor was introduced to vary the reference velometer readings . This correction factor was also included into the computer data acquisition program and its determination can also be found later in this Appendix {pg77 1700 rpm, pg 87 1150 rpm, pg 97 2300 rpm}.

The procedures presented here for obtaining the duct normalized velocity profiles were carried out for fan shaft speeds of 1150, 1700, and 2300 RPM which yielded approximate coil face velocities of 1.3, 2.2, and 3.1 m/s respectively.

DUCT VELOCITY PROFILE

DATE 17/8/84
DAMPER # 5
FAN SPEED 1700 RPM
AMBIENT TEMP. 23.4 Deg.C
BAROMETRIC PRESSURE 29.434 in.Hg

ALL READINGS TAKEN FROM VELOMETER IN FPM!

DOWNSTREAM EVAPORATOR

LOCATION #	FPM	LOCATION #	FPM	LOCATION #	FPM
0	450	1	470	2	465
3	455	4	440	5	440
6	450	7	435	8	430
9	430	10	415	11	370
12	410	13	395	14	365
15	220	16	270	17	360

UPSTREAM EVAPORATOR

LOCATION #	FPM	LOCATION #	FPM	LOCATION #	FPM
20	440	21	440	22	395
23	415	24	380	25	445
26	440	27	405	28	440
29	355	30	435	31	455
32	350	33	390	34	400
35	405	36	430	37	465

DOWNSTREAM CONDENSER

LOCATION #	FPM	LOCATION #	FPM	LOCATION #	FPM
40	355	41	430	42	525
43	340	44	440	45	535
46	425	47	430	48	515
49	450	50	450	51	510
52	375	53	420	54	510
55	335	56	380	57	430

UPSTREAM CONDENSER

LOCATION #	FPM	LOCATION #	FPM	LOCATION #	FPM
60	380	61	360	62	400
63	520	64	380	65	375
66	470	67	345	68	360
69	580	70	340	71	360
72	410	73	300	74	330
75	510	76	350	77	325

TABLE A.1
ORIGINAL DUCT VELOCITY DISTRIBUTION FOR FAN SPEED OF 1700 RPM

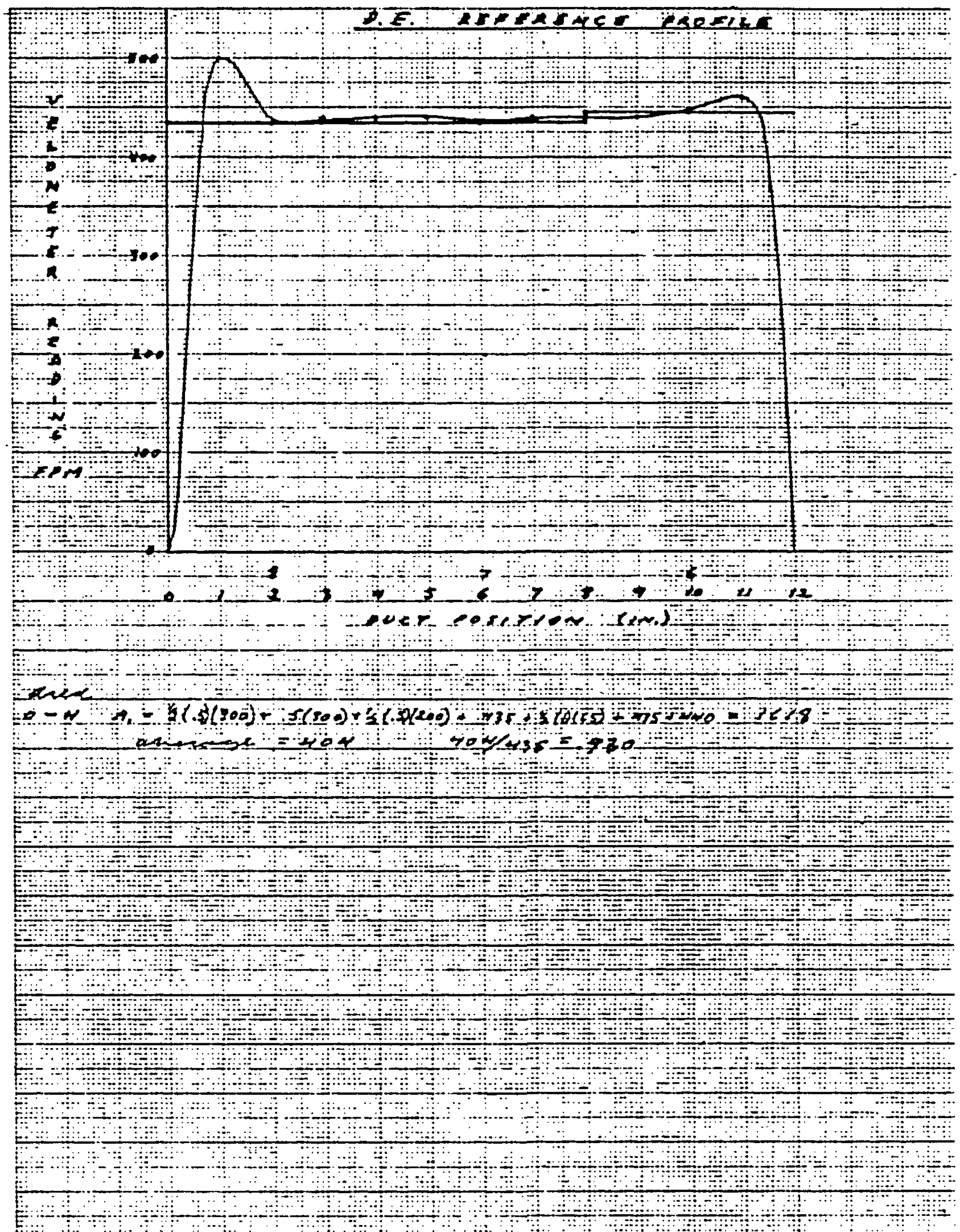
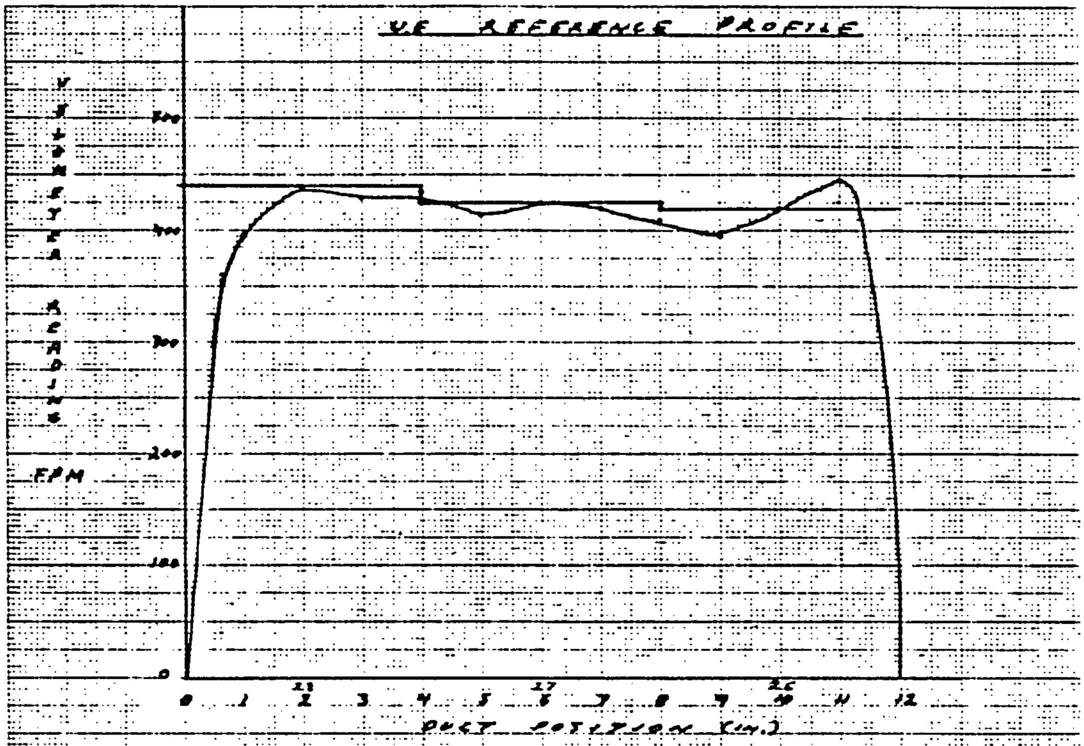


FIGURE A.2
DOWNSTREAM EVAPORATOR DETAILED VELOCITY DISTRIBUTION



Area

0-4 $A_1 = (5)\frac{1}{2}(300) + .5(100) + \frac{1}{2}(5)95 + 395 + \frac{1}{2}(45) + 430 + \frac{1}{2}(10) + 430 + \frac{1}{2}(5) = 1533.75$

average $= 383.4$ $383.4/440 = .871$

8-12 $A_2 = 395 + \frac{1}{2}(15) + 395 + \frac{1}{2}(25) + 420 + \frac{1}{2}(25) + 365(.5) + \frac{1}{2}(.5)(50) + \frac{1}{2}(.5)(365)$

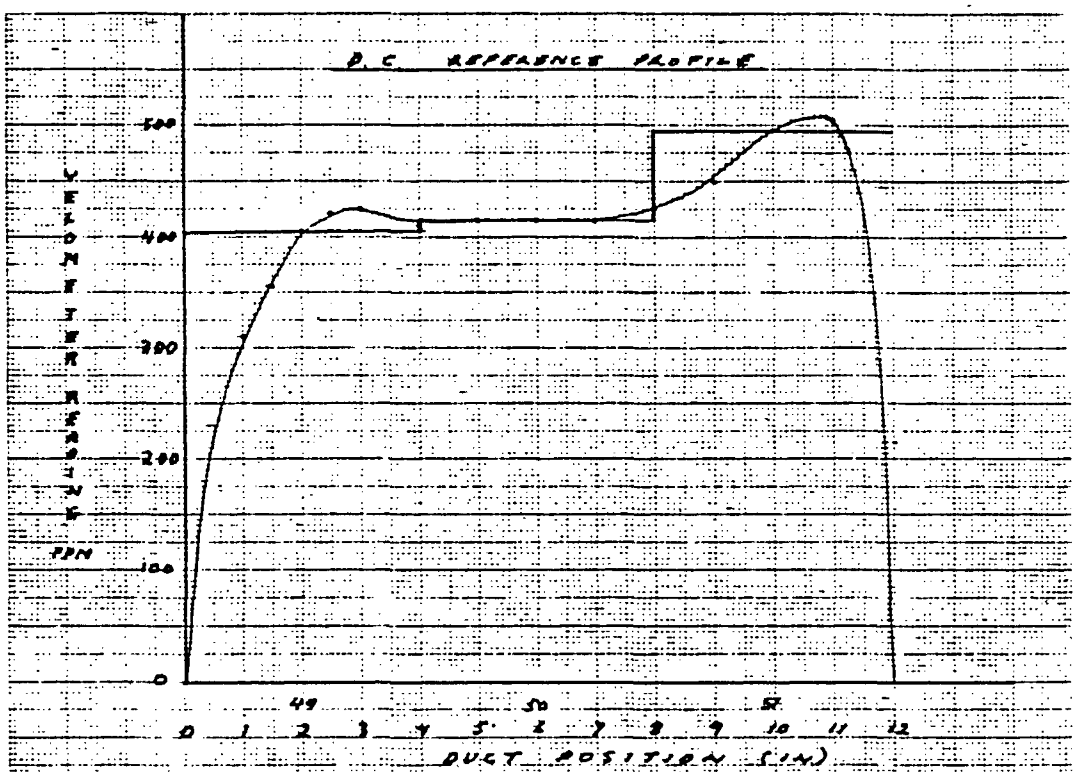
$= 1536.25$

$= 384.1$ $384.1/420 = .914$

4-8 $A_3 = 415 + \frac{1}{2}(20) + 415 + \frac{1}{2}(10) + 420 + \frac{1}{2}(5) + 410 + \frac{1}{2}(20) = 1682.5$

average $= 420.6$ $\frac{420.6}{425}$

FIGURE A.3
UPSTREAM EVAPORATOR DETAILED VELOCITY DISTRIBUTION



AREA

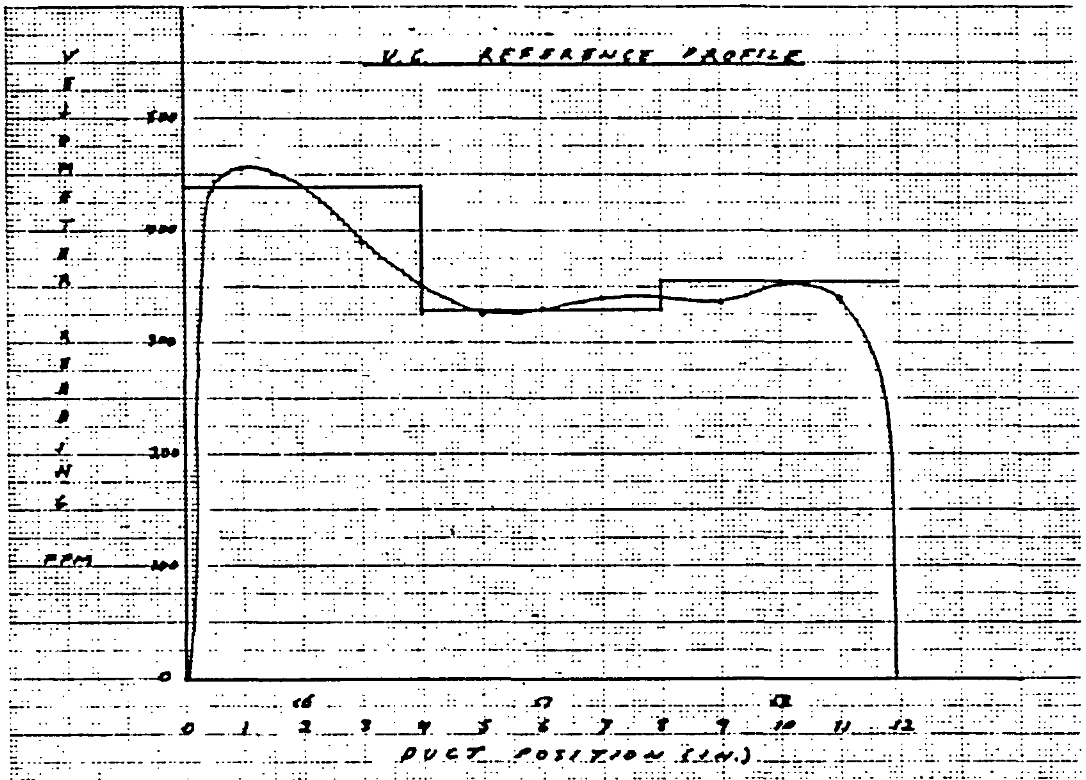
0-4 $A_1 = \frac{1}{2}(1)310 + 310(.5) + \frac{1}{2}(1.0)(45) + 355(.5) + \frac{1}{2}(.5)(60) + 415(.5) + \frac{1}{2}(.5)(15) + 420(.5) = 415 + \frac{1}{2}(1.0)(10)$
$= 1345$

average $= 336.25$ $\frac{336.25}{405} = .830$

4-12 $A_2 = 425 + \frac{1}{2}(25) + 450 + \frac{1}{2}(45) + 495 + \frac{1}{2}(10) + \frac{1}{2}(505) = 1282.5$

average $= 415.6$ $\frac{415.6}{495} = .840$

FIGURE A.4
DOWNSTREAM CONDENSER DETAILED VELOCITY DISTRIBUTION



AREA

0 - 4 $A_1 = \frac{1}{2}(.5)(440) + .5(440) + 440 + \frac{1}{2}(15) + 390 + \frac{1}{2}(50) + 350 + \frac{1}{2}(40) + \frac{1}{2}(.5)(15) = 1546.25$

average = 397.6 $\frac{391.6}{440} = .890$

8-12 $A_2 = 335 + 335 + \frac{1}{2}(20) + 340 + \frac{1}{2}(15) + 295(.5) + \frac{1}{2}(.5)(45) + \frac{1}{2}(.5)(295) = 1260$

average = 315 315/355 = .887

4-8 $A_3 = 325 + \frac{1}{2}(25) + 325 + \frac{1}{2}(5) + 330 + \frac{1}{2}(10) + 340 = 1340$

average = 335 335/330 = 1.02

FIGURE A.5
UPSTREAM CONDENSER DETAILED VELOCITY DISTRIBUTION

CORRECTED DUCT VELOCITY PROFILE

DATE 17/8/84
DAMPER # 5
FAN SPEED 1700 RPM
AMBIENT TEMP. 23.4 Deg.C
BAROMETRIC PRESSURE 29.434 in.Hg

ALL READINGS TAKEN FROM VELOMETER IN FPM!

DOWNSTREAM EVAPORATOR

LOCATION #	FPM	LOCATION #	FPM	LOCATION #	FPM
0	450	1	470	2	432
3	455	4	440	5	409
6	450	7	435	8	400
9	430	10	415	11	344
12	410	13	395	14	339
15	220	16	270	17	335

UPSTREAM EVAPORATOR

LOCATION #	FPM	LOCATION #	FPM	LOCATION #	FPM
20	402	21	440	22	344
23	379	24	380	25	388
26	402	27	405	28	383
29	324	30	435	31	396
32	320	33	390	34	348
35	370	36	430	37	405

DOWNSTREAM CONDENSER

LOCATION #	FPM	LOCATION #	FPM	LOCATION #	FPM
40	295	41	430	42	433
43	282	44	440	45	449
46	353	47	430	48	433
49	324	50	450	51	428
52	311	53	420	54	428
55	228	56	380	57	361

UPSTREAM CONDENSER

LOCATION #	FPM	LOCATION #	FPM	LOCATION #	FPM
60	338	61	360	62	355
63	463	64	380	65	333
66	418	67	345	68	319
69	516	70	340	71	319
72	365	73	300	74	293
75	454	76	350	77	288

DUCT VELOCITY NORMALIZED PROFILE

FAN SPEED: 17..

DAMPER POSITION # 5

DATE : 17/8/8-

REFERENCES USED: 7.27.50 67

LOC.#	ORIGINAL FPM	CALC. M/S	RATIO
0	450	2.43168495	1.03064153
1	470	2.52807867	1.07149685
2	433	2.34493064	.993871695
3	455	2.45578777	1.04085574
4	440	2.38348811	1.01021384
5	409	2.23407791	.946888052
6	450	2.43168495	1.03064151
7	435	2.35938969	1
8	400	2.19070075	.928503148
9	430	2.33529127	.989786164
10	415	2.26299601	.959144656
11	344	1.92079846	.814108184
12	410	2.23889759	.94893082
13	395	2.16660234	.918289312
14	339	1.89670004	.803894348
15	220	1.32315766	.56080505
16	270	1.56414185	.66294341
17	335	1.8774213	.795733279
20	403	2.20034012	.993471619
21	440	2.38348811	1.07616444
22	344	1.92079846	.867256264
23	379	2.08946739	.943420702
24	380	2.09430708	.94559683
25	388	2.13286455	.963005844
26	403	2.20034012	.993471619
27	405	2.21479918	1
28	383	2.10676613	.95121251
29	324	1.82440478	.823733727
30	435	2.35938969	1.06528381
31	396	2.17142202	.980414858
32	320	1.80512604	.81502922
33	390	2.14250392	.967358397
34	348	1.94007719	.87596077
35	370	2.04611024	.923835561
36	430	2.33529127	1.05440317
37	405	2.2147991B	1
40	295	1.68463795	.692784625
41	430	2.33529127	.960359307
42	433	2.34975032	.966305411
43	282	1.62197806	.667018175
44	440	2.38348811	.980179654
45	449	2.42686526	.998017965
46	357	1.96417561	.807742637
47	430	2.33529127	.960359307
48	433	2.34975032	.966305411
49	374	2.06538897	.849365365
50	450	2.43168495	1
51	428	2.3256519	.956395237
52	311	1.76174889	.72449718
53	420	2.28709443	.94053896
54	428	2.3256519	.956395237
55	278	1.60269932	.65909036
56	380	2.09430708	.861257573
57	361	2.00273308	.823598914
60	338	1.89188035	.982479503
61	360	1.9979134	1.03754792
62	355	1.97381498	1.02502928
63	463	2.49434084	1.29534557
64	380	2.09430708	1.08760249
65	333	1.86778193	.969964862
66	418	2.27745507	1.18271376
67	345	1.92561814	1
68	319	1.80030636	.934923867
69	516	2.74978409	1.42800072
70	340	1.90151972	.987485359
71	319	1.80030636	.934923867
72	365	2.02201162	1.05005856
73	300	1.70673337	.886368233
74	293	1.67499458	.869847735
75	454	2.45096368	1.27281917
76	350	1.94971656	1.01251464
77	296	1.65089616	.857333094

1700 Correction factor determination

Profile before correction factor

LOC.#	RATIO	CALC(M/S)
0	1.0306	2.50609451
1	1.0715	2.60555042
2	.9939	2.41685167
3	1.0409	2.53114087
4	1.0102	2.45648814
5	.9469	2.30256248
6	1.0306	2.50609451
7	1	2.43168495
8	.9285	2.25781947
9	.9898	2.40688176
10	.9591	2.33222902
11	.8141	1.97963472
12	.9489	2.30742585
13	.9183	2.23301629
14	.8039	1.95483153
15	.5608	1.36368893
16	.6629	1.61196395
17	.7957	1.93489171

REFERENCE = 450
AVERAGE VELOCITY (M/S)= 2.22993616

20	.9935	2.36799544
21	1.0762	2.56510991
22	.8673	2.06719924
23	.9434	2.24858268
24	.9456	2.25382636
25	.963	2.29529905
26	.9935	2.36799544
27	1	2.38348811
28	.9521	2.26931903
29	.8237	1.96327916
30	1.0653	2.53912989
31	.9804	2.33677174
32	.815	1.94254281
33	.9674	2.3057864
34	.876	2.08793558
35	.9238	2.20186633
36	1.0544	2.51314986
37	1	2.38348811

REFERENCE = 440
AVERAGE VELOCITY (M/S)= 2.2829314

40	.6928	1.66797595
41	.9604	2.3122461
42	.9663	2.32645086
43	.667	1.60586022
44	.9802	2.35991632
45	.998	2.40277136
46	.8077	1.94460764
47	.9604	2.3122461
48	.9663	2.32645086
49	.8494	2.045004
50	1	2.40758653
51	.9564	2.30261576
52	.7245	1.74429644
53	.9405	2.26433513
54	.9564	2.30261576
55	.6591	1.58684028
56	.8613	2.07365428
57	.8236	1.98288827

REFERENCE = 445
AVERAGE VELOCITY (M/S)= 2.10935344

60	.9825	1.96294991
61	1.0375	2.07283515
62	1.025	2.04786123
63	1.2953	2.58789723
64	1.0876	2.17293061
65	.97	1.937976
66	1.1827	2.36293218
67	1	1.9979134
68	.9349	1.86784924
69	1.428	2.85302033
70	.9875	1.97293948
71	.9349	1.86784924
72	1.05	2.09780907
73	.8874	1.77294835
74	.8698	1.73778508
75	1.2728	2.54294418
76	1.0125	2.02288732
77	.8573	1.71281116

REFERENCE = 360
AVERAGE VELOCITY (M/S)= 2.08845218

AVER VEL. FPM	REF VEL
408.140404	450
419.135981	440
383.121618	445
378.784974	360

TOTAL AVERAGE:397.295744

1700
CORRECTION FACTOR DETERMINATION
for average velocity

LOC.#	RATIO	CALC(M/S)
0	1.0306	2.44648851
1	1.0715	2.54357893
2	.9939	2.35936827
3	1.0409	2.47093916
4	1.0102	2.398062
5	.9469	2.24779737
6	1.0306	2.44648851
7	1	2.37384874
8	.9285	2.20411856
9	.9898	2.34963549
10	.9591	2.27675833
11	.8141	1.93255026
12	.9489	2.25254507
13	.9183	2.1799053
14	.8039	1.908337
15	.5608	1.33125438
16	.6629	1.57362433
17	.7957	1.88887144

REFERENCE = 438
AVERAGE VELOCITY (M/S)= 2.17689843

20	.9935	2.25786325
21	1.0762	2.4458102
22	.8673	1.97105667
23	.9434	2.14400422
24	.9456	2.14900402
25	.963	2.18854787
26	.9935	2.25786325
27	1	2.27263538
28	.9521	2.16377615
29	.8237	1.87196976
30	1.0653	2.42103847
31	.9804	2.22809173
32	.815	1.85219784
33	.9674	2.19854747
34	.876	1.99082859
35	.9238	2.09946056
36	1.0544	2.39626675
37	1	2.27263538

REFERENCE = 417
AVERAGE VELOCITY (M/S)= 2.17675542

40	.6928	1.72140118
41	.9604	2.38630729
42	.9663	2.40096703
43	.667	1.65729588
44	.9802	2.43550438
45	.998	2.47973207
46	.8077	2.00689338
47	.9604	2.38630729
48	.9663	2.40096703
49	.8494	2.11050543
50	1	2.48470147
51	.9564	2.37636849
52	.7245	1.80016622
53	.9405	2.33686173
54	.9564	2.37636849
55	.6591	1.63766674
56	.8613	2.14007338
57	.8236	2.04640013

REFERENCE = 461
AVERAGE VELOCITY (M/S)= 2.17691598

60	.9825	2.04818602
61	1.0375	2.16284275
62	1.025	2.1367844
63	1.2953	2.70027008
64	1.0876	2.2672846
65	.97	2.02212768
66	1.1827	2.4655365
67	1	2.08466771
68	.9349	1.94895584
69	1.428	2.97690549
70	.9875	2.0586093c
71	.9349	1.94895584
72	1.05	2.18890109
73	.8874	1.84993413
74	.8698	1.81324397
75	1.2728	2.65336506
76	1.0125	2.11072606
77	.8573	1.78718563

REFERENCE = 378
AVERAGE VELOCITY (M/S)= 2.1791379

∴ correction should be:

	AVE VEL	NEW REF
DE	397.136011	438
VE	397.10634	417
Dc	397.139653	461
Vc	397.600662	378

TOTAL AVERAGE:397.245666
PR#0

438/450 = .973 DE
417/440 = .948 VE
461/445 = 1.036 OC
378/360 = 1.050 VC

CHECKED DUCT VELOCITY PROFILE

DATE 2/15/85
DAMPER # 5
FAN SPEED 1700 RPM
AMBIENT TEMP. 16.1 Deg.C
BAROMETRIC PRESSURE 29.52 in.Hg

ALL READINGS TAKEN FROM VELOMETER IN FPM!

DOWNSTREAM EVAPORATOR

ave	LOCATION #	FPM	LOCATION #	FPM	LOCATION #	FPM	Largest difference (FPM)	AVE DIFF
	0	450	1	465	2	460		
	3	440	4	420	5	420		
	6	455	7	425	8	420		
400.8	9	430	10	410	11	390	20	
	12	410	13	400	14	370		
	15	220	16	280	17	350		

UPSTREAM EVAPORATOR

	LOCATION #	FPM	LOCATION #	FPM	LOCATION #	FPM	
	20	450	21	440	22	410	
	23	415	24	380	25	460	
424.2	26	440	27	415	28	450	
	29	375	30	435	31	465	20
	32	360	33	400	34	410	
	35	410	36	450	37	420	

DOWNSTREAM CONDENSER

	LOCATION #	FPM	LOCATION #	FPM	LOCATION #	FPM	
	40	400	41	440	42	500	
	43	340	44	420	45	510	
438.1	46	450	47	450	48	500	45
	49	460	50	450	51	510	
	52	375	53	400	54	505	
	55	355	56	390	57	430	

UPSTREAM CONDENSER

	LOCATION #	FPM	LOCATION #	FPM	LOCATION #	FPM	
	60	410	61	350	62	400	
	63	500	64	380	65	385	
399.2	66	470	67	350	68	380	35
	69	590	70	350	71	360	
	72	430	73	320	74	340	
	75	425	76	365	77	330	

DUCT VELOCITY PROFILE

DATE 20/7/84
DAMPER # 5
FAN SPEED 1150 RPM
AMBIENT TEMP. 25.9 Deg.C

ALL READINGS TAKEN FROM VELOMETER IN FPM!

DOWNSTREAM EVAPORATOR

LOCATION #	FPM	LOCATION #	FPM	LOCATION #	FPM
0	270	1	285	2	280
3	255	4	250	5	250
6	290	7	265	8	250
9	240	10	240	11	220
12	230	13	230	14	210
15	125	16	150	17	200

UPSTREAM EVAPORATOR

LOCATION #	FPM	LOCATION #	FPM	LOCATION #	FPM
20	225	21	240	22	210
23	225	24	210	25	260
26	250	27	225	28	255
29	215	30	225	31	245
32	195	33	220	34	280
35	225	36	260	37	265

DOWNSTREAM CONDENSER

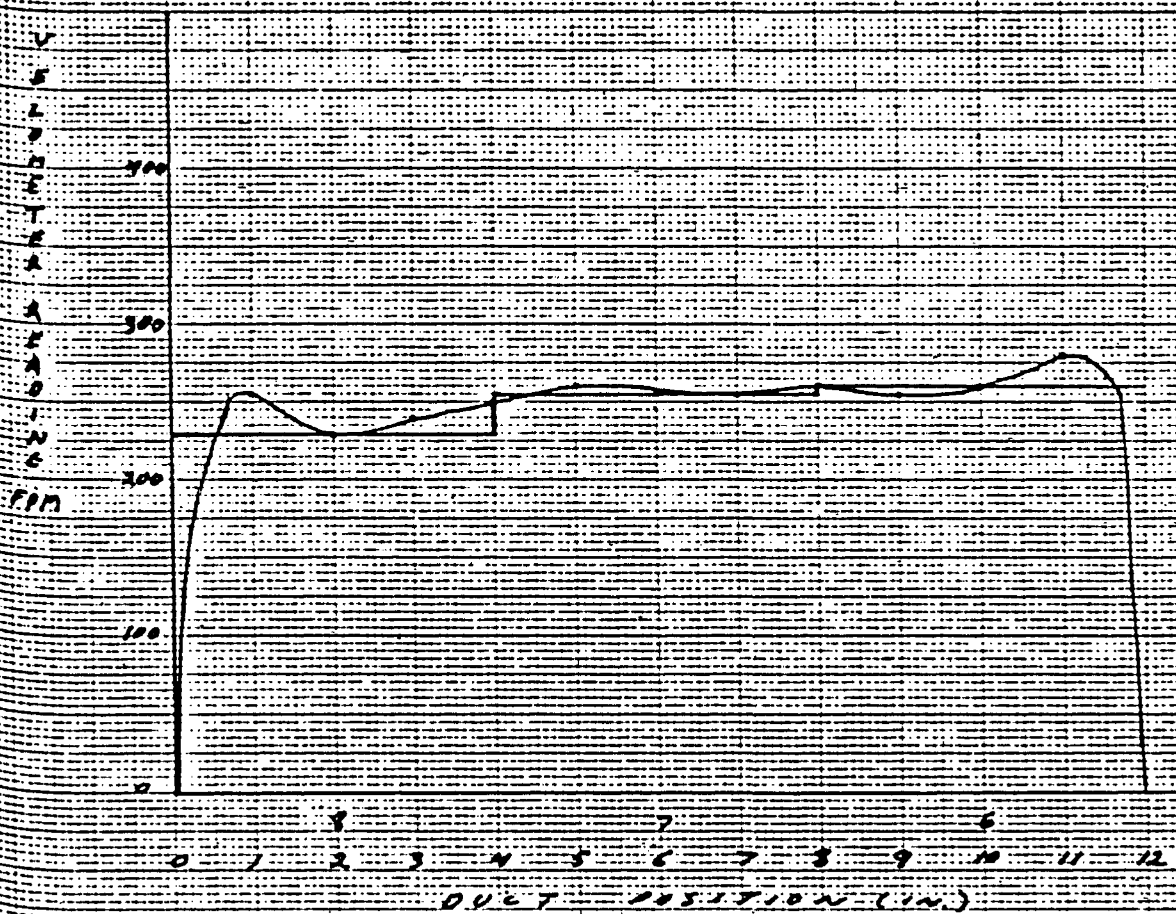
LOCATION #	FPM	LOCATION #	FPM	LOCATION #	FPM
40	225	41	~~260~~ 255	42	260
43	200	44	265	45	310
46	230	47	260	48	315
49	260	50	265	51	285
52	225	53	250	54	320
55	230	56	240	57	265

UPSTREAM CONDENSER

LOCATION #	FPM	LOCATION #	FPM	LOCATION #	FPM
60	170	61	150	62	210
63	290	64	220	65	205
66	260	67	210	68	205
69	285	70	210	71	210
72	260	73	195	74	205
75	240	76	210	77	195

B. E. REFERENCE PROFILE

1150

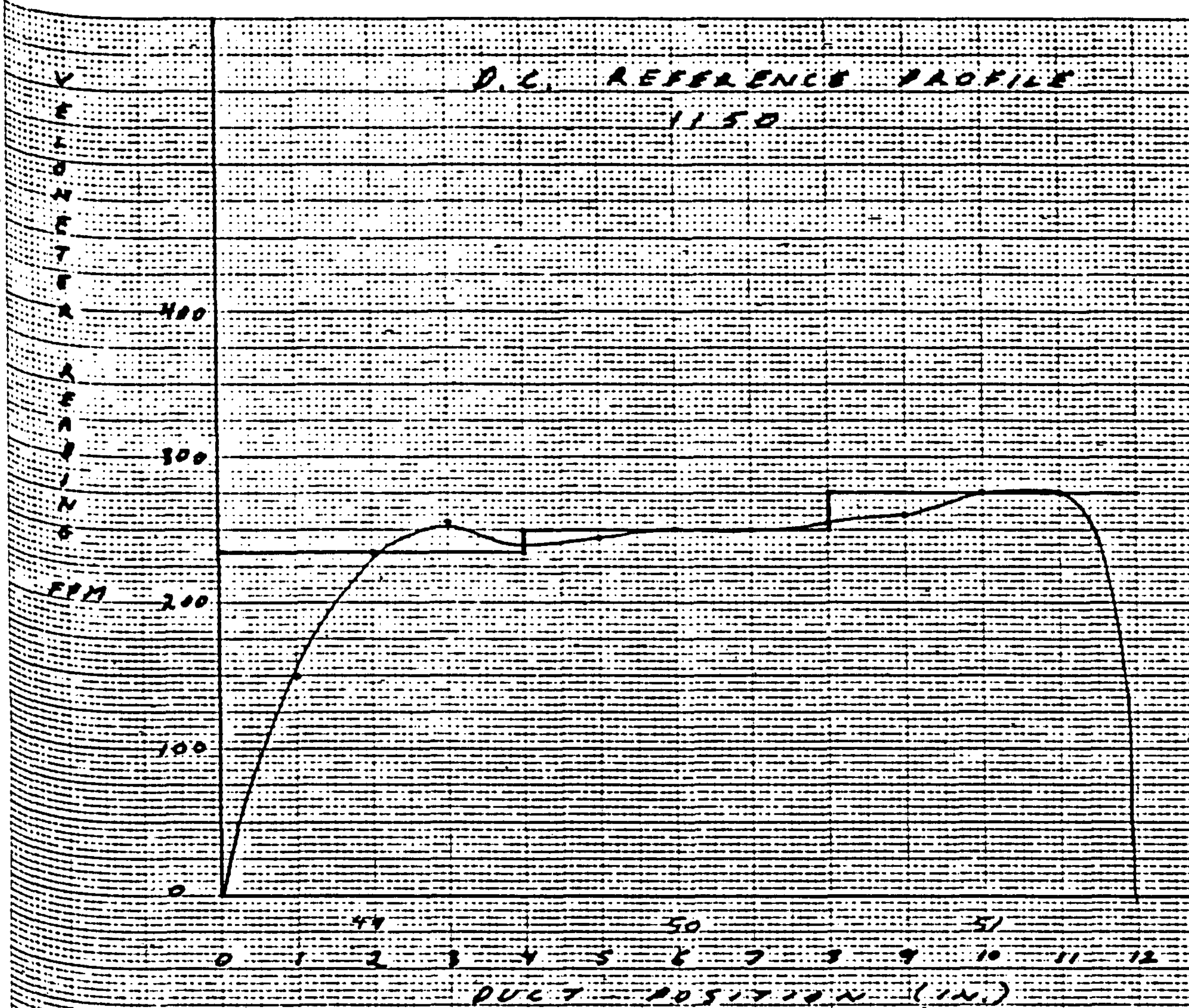


$A_{0-4} = \frac{1}{2}(.5)(220) + .5(220) + \frac{1}{2}(.5)35 + 230 + \frac{1}{2}(25) + 230 + \frac{1}{2}(110) + 240 + \frac{1}{2}(30) = 896.25$

average $= 224.1 \qquad \frac{224.1}{230} = .974$

$A_{0-4} = \tfrac{1}{2}(.5)(190) + (\tfrac{1}{2})90 + \tfrac{1}{2}(.5)(40) + 230 + \tfrac{1}{2}(20) + 250 + 250 = 892.5$

average $= 223.1$ $\qquad 223.1/250 = .8925$



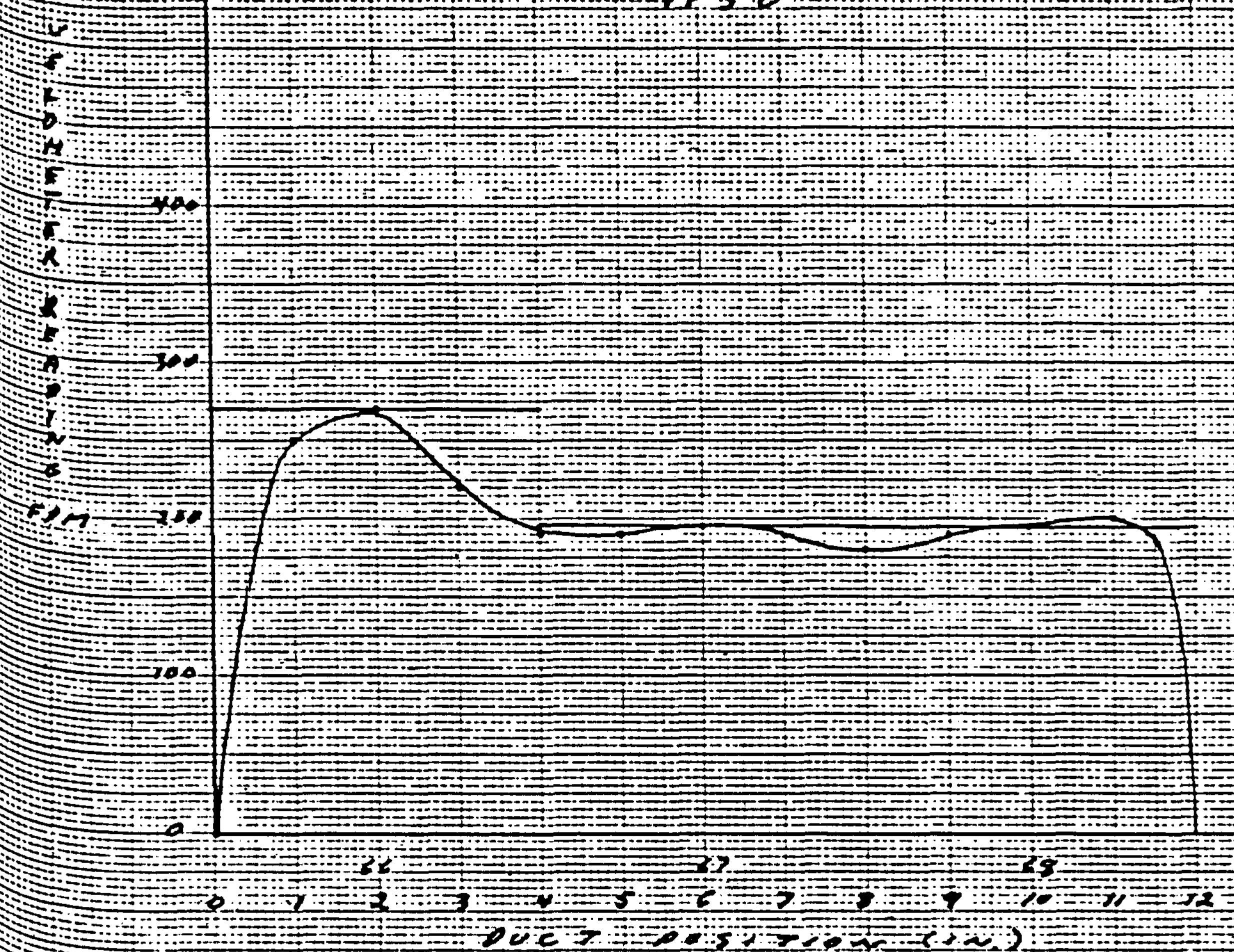
$A_{0-4} = \frac{1}{2}(150) + 150 + \frac{1}{2}(85) + 235 + \frac{1}{2}(20) + 240 + \frac{1}{2}(15) = 760$

average $= 190$ $\quad 190/235 = .8085$

$A_{8-12} = 255 + \frac{1}{2}(5) + 260 + \frac{1}{2}(15) + 275 + \frac{1}{2}(.5)(30) + 245(.5) + \frac{1}{2}(.5)(245) = 901.25$

average $= 247.8$ $\quad 247.8/275 = .901$

V.C. REFERENCE PROFILE
1150



$A_{0-4} = \frac{1}{2}(.5)(175) + 175(.5) + \frac{1}{2}(.5)(75) + 250 + \frac{1}{2}(20) + 220 + \frac{1}{2}(50) + 190 + \frac{1}{2}(30) = 860$

average = 215   215/220 = .796

CORRECTED DUCT VELOCITY PROFILE

DATE 20/7/84
DAMPER # 5
FAN SPEED 1150 RPM
AMBIENT TEMP. 25.9 Deg.C
BAROMETRIC PRESSURE / in.Hg

ALL READINGS TAKEN FROM VELOMETER IN FPM!

DOWNSTREAM EVAPORATOR

LOCATION #	FPM	LOCATION #	FPM	LOCATION #	FPM
0	270	1	285	2	273
3	255	4	250	5	244
6	290	7	265	8	244
9	240	10	240	11	214
12	230	13	230	14	205
15	125	16	150	17	195

UPSTREAM EVAPORATOR

LOCATION #	FPM	LOCATION #	FPM	LOCATION #	FPM
20	225	21	240	22	197
23	225	24	210	25	232
26	250	27	225	28	228
29	215	30	225	31	219
32	195	33	220	34	197
35	225	36	260	37	237

DOWNSTREAM CONDENSER

LOCATION #	FPM	LOCATION #	FPM	LOCATION #	FPM
40	182	41	255	42	234
43	162	44	265	45	279
46	186	47	260	48	294
49	210	50	265	51	257
52	182	53	250	54	289
55	186	56	240	57	239

UPSTREAM CONDENSER

LOCATION #	FPM	LOCATION #	FPM	LOCATION #	FPM
60	135	61	150	62	210
63	231	64	220	65	205
66	207	67	210	68	205
69	227	70	210	71	210
72	207	73	195	74	205
75	191	76	210	77	195

DUCT VELOCITY NORMALIZED PROFILE

FAN SPEED:115

DAMPER POSITION # 5

DATE :20/7/84

REFERENCES USED: 7.17.50.67

LOC.#	ORIGINAL FPM	CALC. M/S	RATIO
0	270	1.56414185	1.01564785
1	285	1.63643711	1.06259153
2	273	1.5786009	1.0250366
3	255	1.49184659	.968704234
4	250	1.46774817	.95305635
5	244	1.43883007	.934278891
6	290	1.66053553	1.07823942
7	265	1.54004343	1
8	244	1.43883007	.93427889
9	240	1.41955133	.921760585
10	240	1.41955133	.921760585
11	214	1.29423955	.840391593
12	230	1.37135449	.890464819
13	230	1.37135449	.890464819
14	205	1.2508624	.812225404
15	125	.865287685	.561859275
16	150	.985779782	.640098569
17	195	1.20266556	.780929638
20	225	1.34725607	1
21	240	1.41955133	1.05366111
22	187	1.16410809	.864058518
23	225	1.34725607	1
24	210	1.27496082	.946338888
25	232	1.38099386	1.02504185
26	250	1.46774817	1.08943519
27	225	1.34725607	1
28	228	1.36171513	1.01073222
29	215	1.29905923	.964225926
30	225	1.34725607	1
31	219	1.31833797	.978535556
32	195	1.20266556	.892677777
33	220	1.32315766	.982112963
34	187	1.16410809	.864058518
35	225	1.34725607	1
36	260	1.51594501	1.12520926
37	237	1.40509228	1.04292889
40	182	1.14000967	.740245142
41	255	1.49184659	.968704234
42	234	1.39063323	.902983125
43	162	1.04361599	.67765361
44	265	1.54004343	1
45	279	1.607519	1.04381407
46	186	1.1592884	.752763448
47	260	1.51594501	.984352117
48	284	1.63161742	1.05946196
49	210	1.27496082	.827873287
50	265	1.54004343	1
51	257	1.50148596	.974963387
52	182	1.14000967	.740245142
53	250	1.46774817	.953056351
54	285	1.65089616	1.07198026
55	186	1.1592884	.752763448
56	240	1.41955133	.921760585
57	239	1.41473165	.918631008
60	135	.913484524	.71648047
61	150	.985779782	.773184376
62	210	1.27496082	1
63	231	1.37617418	1.07938547
64	220	1.32315766	1.0378026
65	205	1.2508624	.981098698
66	207	1.26050176	.988659219
67	210	1.27496082	1
68	205	1.2508624	.981098698
69	227	1.35689544	1.06426443
70	210	1.27496082	1
71	210	1.27496082	1
72	207	1.26050176	.988659219
73	195	1.20266556	.943296094
74	205	1.2508624	.961098698
75	191	1.18338682	.928175052
76	210	1.27496082	1
77	195	1.20266556	.943296094

1150

LOC.#	RATIO	CALC(M/S)
0	1.0156	-1.5151194
1	1.0626	1.58523619
2	1.025	1.52914276
3	.9687	1.44515179
4	.9531	1.42187899
5	.9343	1.39383227
6	1.0782	1.60850899
7	1	1.49184659
8	.9343	1.39383227
9	.9218	1.37518419
10	.9218	1.37518419
11	.8404	1.25374788
12	.8905	1.32848939
13	.8905	1.32848939
14	.8122	1.2116778
15	.5619	.838268599
16	.6401	.954931003
17	.7809	1.164983

REFERENCE = 255
AVERAGE VELOCITY (M/S)= 1.34530582

20	1	1.41955133
21	1.0537	1.49578124
22	.8641	1.22663431
23	1	1.41955133
24	.9463	1.34332143
25	1.025	1.45504012
26	1.0894	1.54645922
27	1	1.41955133
28	1.0107	1.43474053
29	.9642	1.3687314
30	1	1.41955133
31	.9785	1.38903098
32	.8927	1.26723347
33	.9821	1.39414136
34	.8641	1.22663431
35	1	1.41955133
36	1.1252	1.59727916
37	1.0429	1.48045009

REFERENCE = 240
AVERAGE VELOCITY (M/S)= 1.40684635

40	.7402	1.06858955
41	.9687	1.39846351
42	.903	1.30361573
43	.6777	.978361437
44	1	1.44364975
45	1.0438	1.50688161
46	.7528	1.08677953
47	.9844	1.42112882
48	1.0595	1.52954691
49	.8279	1.19519763
50	1	1.44364975
51	.975	1.40755851
52	.7402	1.06858955
53	.9531	1.37594258
54	1.072	1.54759253
55	.7528	1.08677953
56	.9218	1.33075634
57	.9186	1.32613666

REFERENCE = 245
AVERAGE VELOCITY (M/S)= 1.30662333

60	.7165	.844443355
61	.7732	.911268111
62	1	1.17856714
63	1.0794	1.27214537
64	1.0378	1.22311698
65	.9811	1.15629222
66	.9887	1.16524933
67	1	1.17856714
68	.9811	1.15629222
69	1.0643	1.25434901
70	1	1.17856714
71	1	1.17856714
72	.9887	1.16524933
73	.9433	1.11174238
74	.9811	1.15629222
75	.9282	1.09394602
76	1	1.17856714
77	.9433	1.11174238

REFERENCE = 190
AVERAGE VELOCITY (M/S)= 1.13972026

224.595216	255
237.36379	240
216.569282	245
181.939839	190

AVERAGE OF ABOVE= 215.117032

1150 correction factor determination

LOC.#	RATIO	CALC(M/S)
0	1.0156	1.46617069
1	1.0626	1.53402223
2	1.025	1.479741
3	.9687	1.39846351
4	.9531	1.37594258
5	.9343	1.34880196
6	1.0782	1.55654316
7	1	1.44364975
8	.9343	1.34880196
9	.9218	1.33075634
10	.9218	1.33075634
11	.8404	1.21324325
12	.8905	1.2855701
13	.8905	1.2855701
14	.8122	1.17253233
15	.5619	.811186796
16	.6491	.924080206
17	.7809	1.12734609

REFERENCE = 245
AVERAGE VELOCITY (M/S)= 1.30184325

20	1	1.31351829
21	1.0537	1.38405422
22	.8641	1.13501115
23	1	1.31351829
24	.9463	1.24298236
25	1.025	1.34635624
26	1.0894	1.43094682
27	1	1.31351829
28	1.0107	1.32757293
29	.9642	1.26649433
30	1	1.31351829
31	.9785	1.28527764
32	.8927	1.17257777
33	.9821	1.29000631
34	.8641	1.13501115
35	1	1.31351829
36	1.1252	1.47797078
37	1.0429	1.36986822

REFERENCE = 218
AVERAGE VELOCITY (M/S)= 1.3017623

40	.7402	1.06502202
41	.9687	1.39379469
42	.903	1.29926355
43	.6777	.975095137
44	1	1.43883007
45	1.0438	1.50185082
46	.7528	1.08315128
47	.9844	1.41638432
48	1.0595	1.52444046
49	.8279	1.19120741
50	1	1.43883007
51	.975	1.40285932
52	.7402	1.06502202
53	.9531	1.37134894
54	1.072	1.54242583
55	.7528	1.08315128
56	.9218	1.32631356
57	.9186	1.3217093

REFERENCE = 244
AVERAGE VELOCITY (M/S)= 1.30226111

60	.7165	.961855673
61	.7732	1.03797182
62	1	1.34243639
63	1.0794	1.44902584
64	1.0378	1.39318049
65	.9811	1.31706434
66	.9887	1.32726686
67	1	1.34243639
68	.9811	1.31706434
69	1.0643	1.42875505
70	1	1.34243639
71	1	1.34243639
72	.9887	1.32726686
73	.9433	1.26632025
74	.9811	1.31706434
75	.9282	1.24604946
76	1	1.34243639
77	.9433	1.26632025

REFERENCE = 224
AVERAGE VELOCITY (M/S)= 1.2981882

215.577499	245/255 = .961	OE
215.560704	218/240 = .908	UE
215.664199	244/245 = .996	OC
214.819141	224/190 = 1.179	UC

AVERAGE OF ABOVE= 215.405386

CHECKED DUCT VELOCITY PROFILE

DATE 22/5/85
DAMPER # 5
FAN SPEED 1150 RPM
AMBIENT TEMP. 15.7 Deg.C
BAROMETRIC PRESSURE 29.52 in.Hg

ALL READINGS TAKEN FROM VELOMETER IN FPM!

DOWNSTREAM EVAPORATOR

LOCATION #	FPM	LOCATION #	FPM	LOCATION #	FPM	
0	280	1	305	2	300	
3	280	4	290	5	280	ave
6	280	7	270	8	270	250
9	250	10	240	11	240	
12	245	13	240	14	220	
15	130	16	120	17	220	

UPSTREAM EVAPORATOR

LOCATION #	FPM	LOCATION #	FPM	LOCATION #	FPM	
20	245	21	260	22	220	
23	255	24	240	25	270	
26	260	27	250	28	280	
29	230	30	260	31	260	251.9
32	220	33	245	34	240	
35	240	36	280	37	280	

DOWNSTREAM CONDENSER

LOCATION #	FPM	LOCATION #	FPM	LOCATION #	FPM	
40	270	41	260	42	280	
43	230	44	260	45	305	
46	260	47	260	48	305	265.5
49	265	50	275	51	270	
52	240	53	250	54	310	
55	240	56	240	57	260	

UPSTREAM CONDENSER

LOCATION #	FPM	LOCATION #	FPM	LOCATION #	FPM	
60	200	61	200	62	220	
63	290	64	230	65	215	
66	280	67	210	68	215	
69	275	70	220	71	230	233.6
72	270	73	210	74	230	
75	260	76	230	77	220	

DUCT VELOCITY PROFILE

DATE 25/7/84
DAMPER # 5
FAN SPEED 2300 RPM
AMBIENT TEMP. 22.6 Deg.C
BAROMETRIC PRESSURE 29.644 in.Hg

ALL READINGS TAKEN FROM VELOMETER IN FPM!

DOWNSTREAM EVAPORATOR

LOCATION #	FPM	LOCATION #	FPM	LOCATION #	FPM
0	640	1	665	2	690
3	660	4	625	5	600
6	695	7	645	8	640
9	650	10	615	11	540
12	620	13	610	14	550
15	360	16	445	17	540

UPSTREAM EVAPORATOR

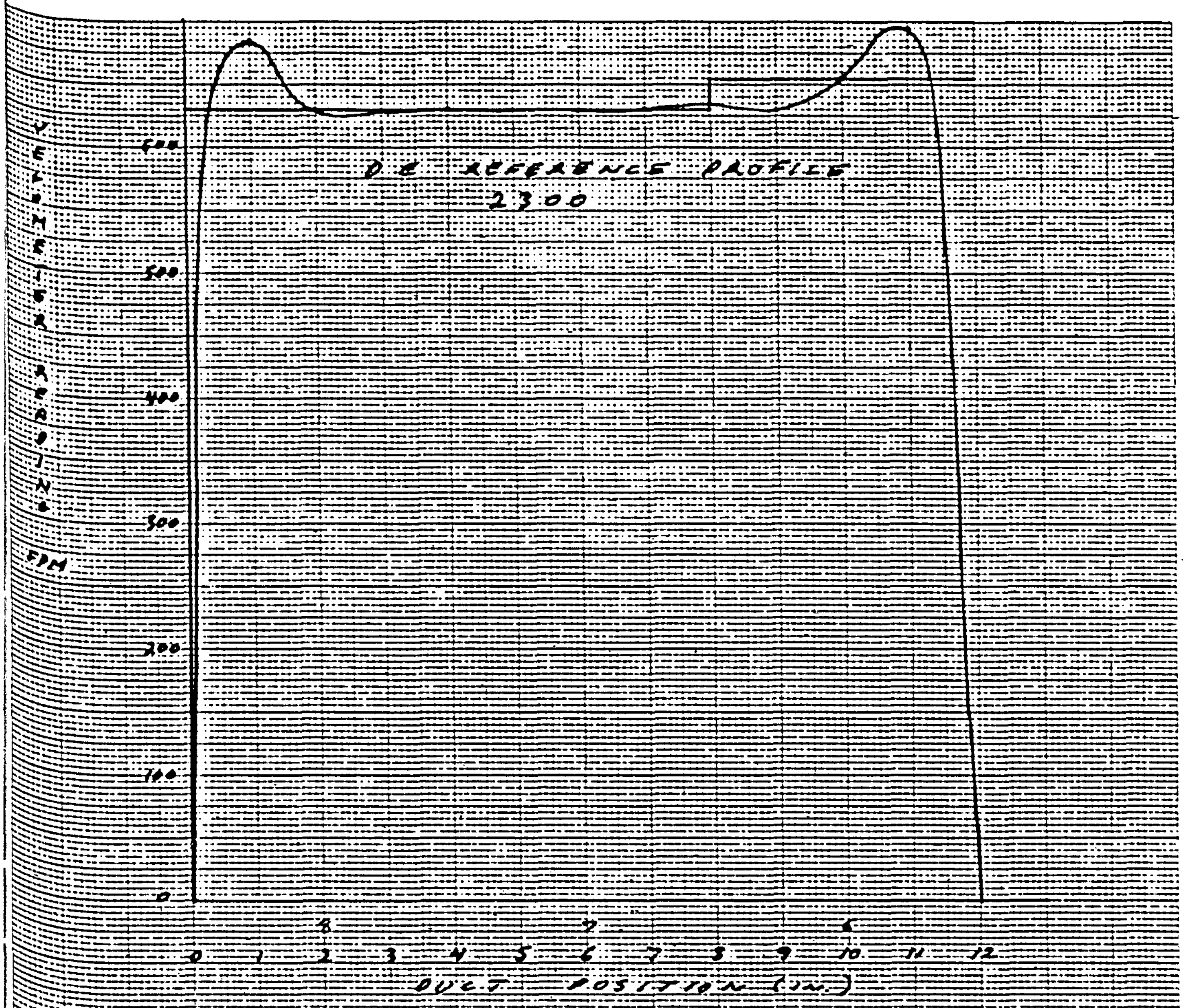
LOCATION #	FPM	LOCATION #	FPM	LOCATION #	FPM
20	645	21	645	22	645
23	570	24	550	25	670
26	640	27	650	28	660
29	550	30	630	31	670
32	530	33	600	34	620
35	570	36	620	37	680

DOWNSTREAM CONDENSER

LOCATION #	FPM	LOCATION #	FPM	LOCATION #	FPM
40	580	41	610	42	750
43	480	44	655	45	800
46	620	47	620	48	760
49	620	50	660	51	830
52	555	53	620	54	765
55	480	56	610	57	660

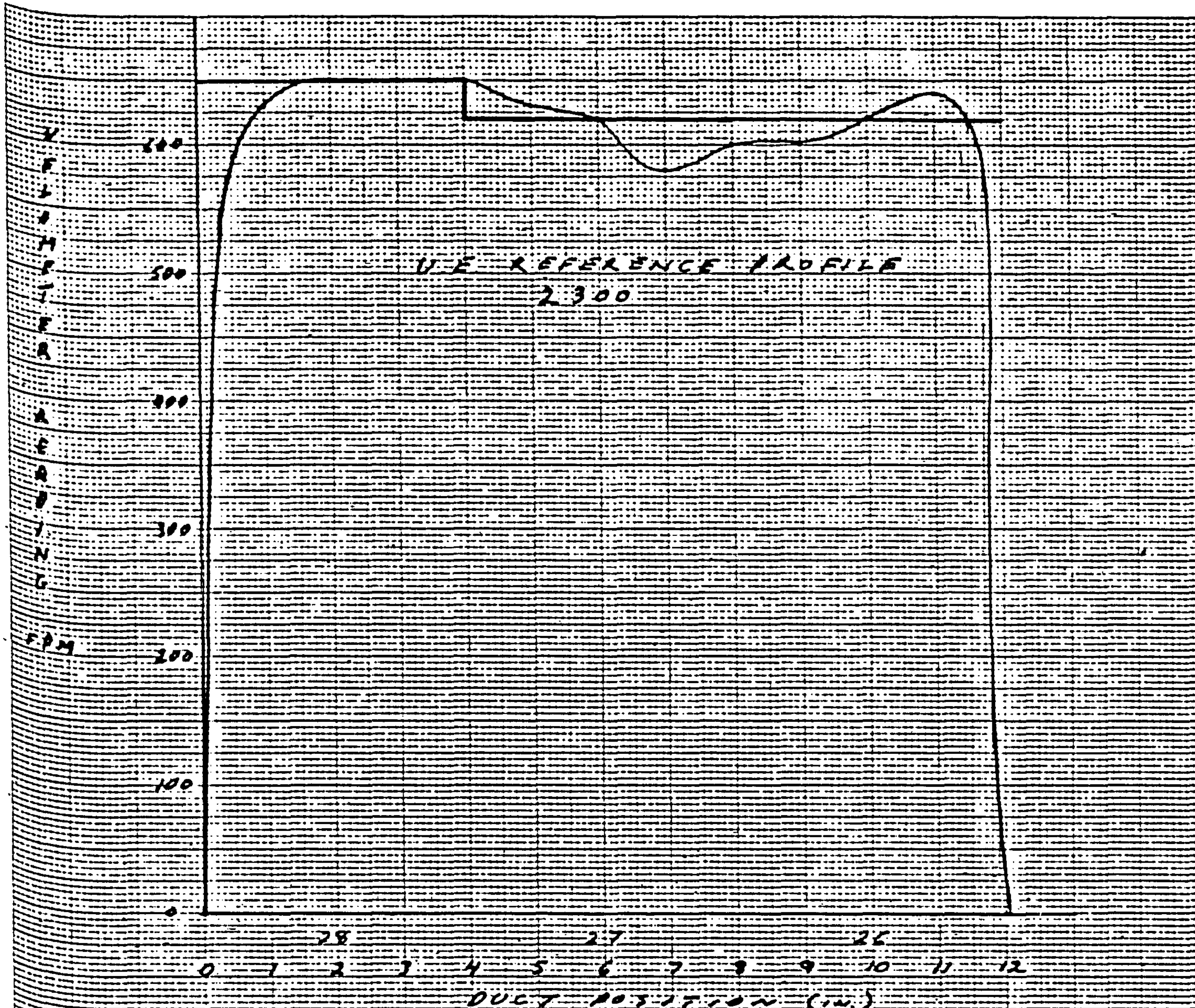
UPSTREAM CONDENSER

LOCATION #	FPM	LOCATION #	FPM	LOCATION #	FPM
60	660	61	560	62	560
63	635	64	590	65	570
66	715	67	590	68	590
69	740	70	560	71	545
72	620	73	510	74	530
75	590	76	550	77	520



$A_{v} = \frac{1}{2}(.2)(550) + 550(.3) + \frac{1}{2}(.3)(105) + 655(.5) + \frac{1}{2}(.5)(30) + 630 + \frac{1}{2}(55) + 630 + 630 = 2488.25$

average $= 622.1$ $\qquad 622.1/630 = .987$



$A_{0-4} = \frac{1}{2}(.5)(585) + .5(585) + \frac{1}{2}(.5)(45) + 630 + \frac{1}{2}(20) + 650 + 650 = 2390$

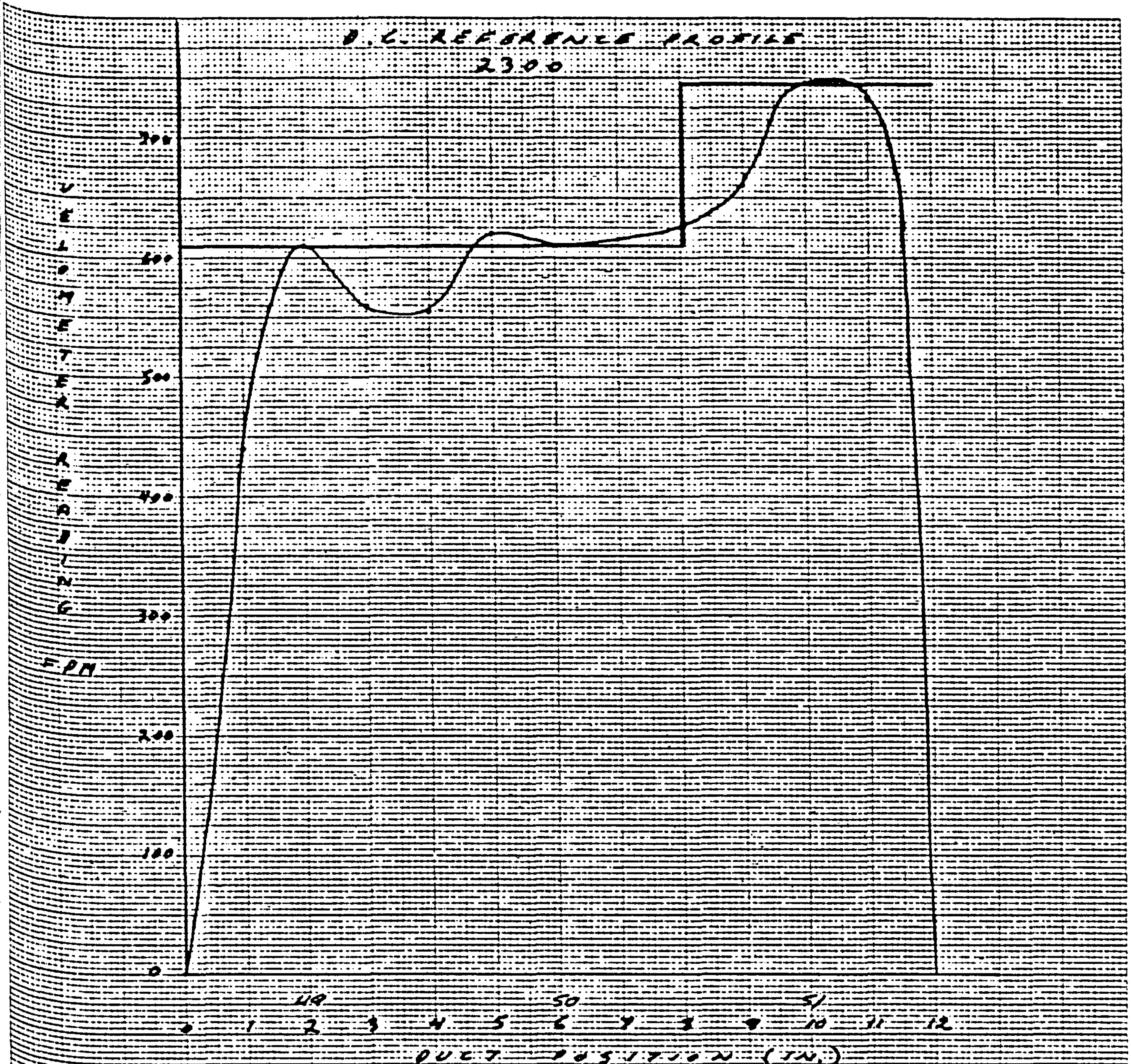
average 597.5 $\qquad$ 597.5/650 = .919

$A_{4-8} = 630 + \frac{1}{2}(20) + 620 + \frac{1}{2}(10) + 580 + \frac{1}{2}(40) + 580 + \frac{1}{2}(20) = 2455$

average = 613.75 $\qquad$ 613.75/620 = .990

$A_{8-12} = 600 + 600 + \frac{1}{2}(20) + 620 + \frac{1}{2}(20) + 620(.5) + \frac{1}{2}(.5)(20) + \frac{2}{3}(.5)(620) = 2310$

average = 577.5 $\qquad$ 577.5/620 = .931



$A_{0-4} = \frac{1}{2}(470) + 470 + \frac{1}{2}(170) + 560 + \frac{1}{2}(50) + 555 + \frac{1}{2}(10) = 1890$

average $= 472.5$ $\quad 472.5/610 = .775$

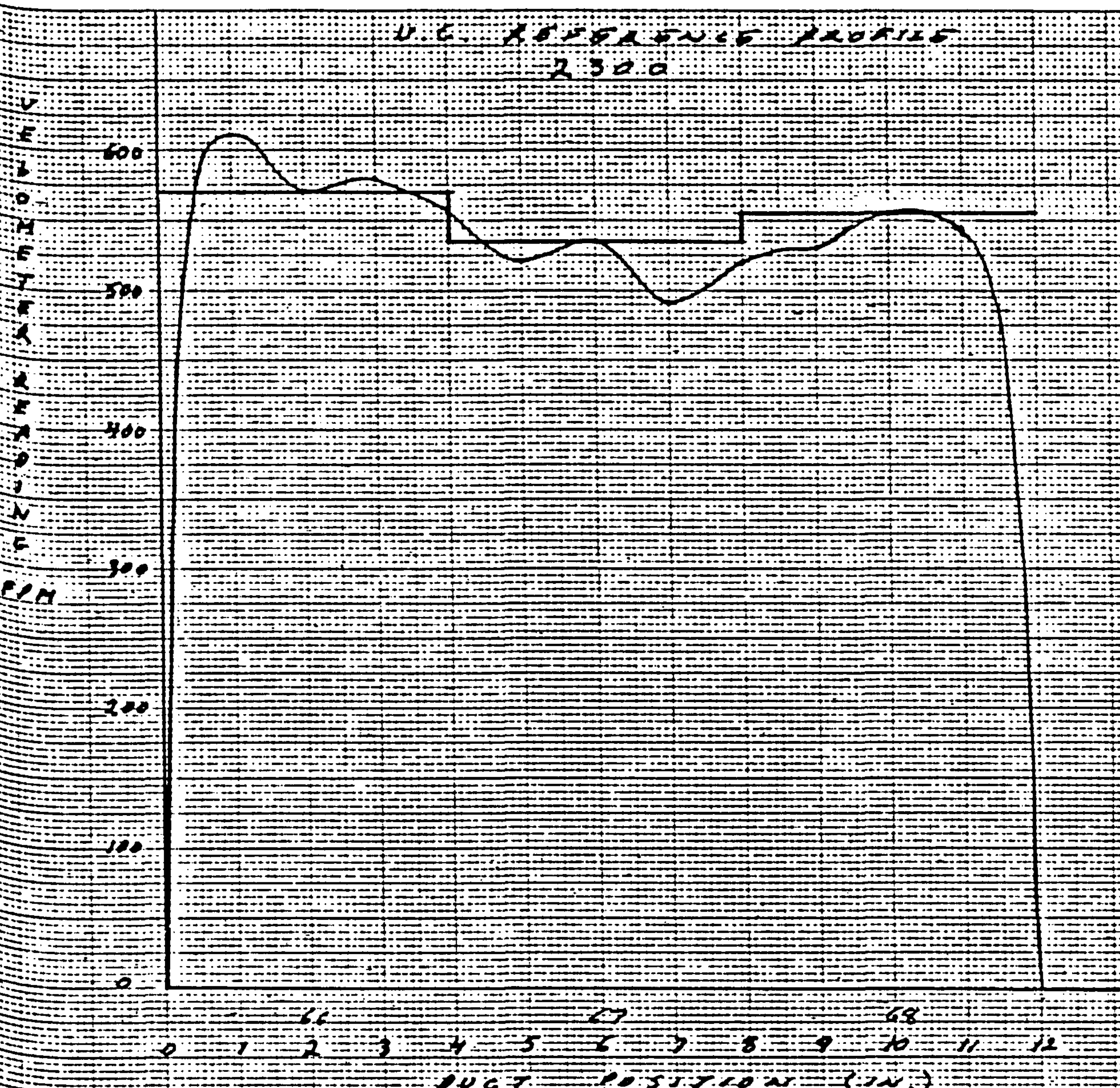
$A_{4-8} = 555 + \frac{1}{2}(65) + 610 + \frac{1}{2}(10) + 610 + \frac{1}{2}(5) + 615 + \frac{1}{2}(0) = 2435$

average $= 608.75$ $\quad 608.75/610 = .998$

$A_{8-12} = 625 + \frac{1}{2}(35) + 660 + \frac{1}{2}(85) + 785 + \frac{1}{2}(10) + 650(.5) + \frac{1}{2}(.5)(65) + \frac{1}{2}(.5)650$

$= 2593.75$

average $= 648.4$ $\quad 648.4/745 = .870$



$A_{0-4} = \frac{1}{2}(.3)(475) + .2(475) + \frac{1}{2}(.2)(95) + 570(.5) + \frac{1}{2}(.5)(40) + 530 + \frac{1}{2}(40) + 570 + \frac{1}{2}(10)$
$+ 555 + \frac{1}{2}(25) = 2203.25$

average $= 550.8$ $\quad 550.8/570 = .966$

$A_{4-8} = 520 + \frac{1}{2}(35) + 520 + \frac{1}{2}(15) + 490 + \frac{1}{2}(45) + 490 + \frac{1}{2}(30) = 2082.5$

average $= 520.6$ $\quad 520.6/535 = .973$

$A_{8-12} = 520 + \frac{1}{2}(10) + 530 + \frac{1}{2}(25) + 545 + \frac{1}{2}(10) + 485(.5) + \frac{1}{2}(.5)(60) + \frac{1}{2}(.5)(485)$
average $= 499.1$ $\quad 499.1/555 = .899$ $\quad = 1996.25$

CORRECTED DUCT VELOCITY PROFILE

DATE 25/7/84
DAMPER # 5
FAN SPEED 2300 RPM
AMBIENT TEMP. 22.6 Deg.C
BAROMETRIC PRESSURE 29.644 in.Hg

ALL READINGS TAKEN FROM VELOMETER IN FPM!

DOWNSTREAM EVAPORATOR

LOCATION #	FPM	LOCATION #	FPM	LOCATION #	FPM
0	640	1	665	2	681
3	660	4	625	5	592
6	695	7	645	8	632
9	650	10	615	11	533
12	620	13	610	14	543
15	360	16	445	17	533

UPSTREAM EVAPORATOR

LOCATION #	FPM	LOCATION #	FPM	LOCATION #	FPM
20	600	21	639	22	543
23	531	24	545	25	616
26	596	27	644	28	607
29	512	30	624	31	616
32	493	33	594	34	576
35	531	36	614	37	625

DOWNSTREAM CONDENSER

LOCATION #	FPM	LOCATION #	FPM	LOCATION #	FPM
40	450	41	610	42	653
43	372	44	655	45	696
46	491	47	620	48	661
49	491	50	660	51	722
52	430	53	620	54	666
55	372	56	610	57	574

UPSTREAM CONDENSER

LOCATION #	FPM	LOCATION #	FPM	LOCATION #	FPM
60	638	61	545	62	503
63	613	64	574	65	512
66	691	67	574	68	530
69	715	70	545	71	490
72	599	73	496	74	476
75	570	76	535	77	467

DUCT VELOCITY NORMALIZED PROFILE

FAN SPEED:2300

DAMPER POSITION # 3

DATE :25/7/84

REFERENCES USED: 7.17.50.67

LOC.#	ORIGINAL FPM	CALC. M/S	RATIO
0	640	3.34742489	.992852364
1	665	3.46791699	1.02859054
2	681	3.54503193	1.05146298
3	660	3.44381857	1.02144291
4	625	3.27512963	.971409456
5	592	3.11608006	.924235065
6	695	3.6125075	1.07147635
7	645	3.37152331	1
8	632	3.30886742	.981416148
9	650	3.39562173	1.00714764
10	615	3.22693279	.957114188
11	533	2.83171871	.839892966
12	620	3.25103121	.964261823
13	610	3.20283437	.949966552
14	543	2.87991555	.854188237
15	360	1.9979134	.59258478
16	445	2.40758653	.714094583
17	533	2.83171871	.839892966
20	600	3.15463753	.937010763
21	639	3.34260521	.992842132
22	593	3.12089975	.926989749
23	531	2.82207934	.838232188
24	545	2.88955492	.858274218
25	616	3.23175248	.959915941
26	596	3.1353588	.931284469
27	644	3.36670362	1
28	607	3.18837532	.947031778
29	512	2.73050535	.81103229
30	624	3.27030995	.971368529
31	616	3.23175248	.959915941
32	493	2.63893136	.783832393
33	594	3.12571943	.928421322
34	576	3.03896512	.902652998
35	531	2.82207934	.838232188
36	614	3.22211311	.957052793
37	625	3.27512963	.972800103
40	450	2.43168495	.706101353
41	610	3.20283437	.930024132
42	653	3.41008078	.990203378
43	372	2.05574961	.596938998
44	655	3.41972015	.993002413
45	696	3.61732719	1.05038263
46	481	2.58109515	.749486391
47	620	3.25103121	.944019305
48	661	3.44863825	1.00139952
49	481	2.58109515	.749486391
50	660	3.44381857	1
51	722	3.74263897	1.08677008
52	430	2.33529127	.678111006
53	620	3.25103121	.944019305
54	666	3.47273667	1.0083971
55	372	2.05574961	.596938998
56	610	3.20283437	.930024132
57	574	3.02932575	.879641507
60	638	3.33778552	1.10182456
61	545	2.88955492	.953860745
62	503	2.6871282	.867028376
63	613	3.21729742	1.06204974
64	574	3.02932575	1
65	512	2.73050535	.901357455
66	691	3.59122877	1.18614803
67	574	3.02932575	1
68	530	2.81725966	.929995613
69	715	3.70890118	1.22433224
70	545	2.88955492	.953860745
71	490	2.6244723	.866355262
72	599	3.14981765	1.03977522
73	496	2.65339041	.875901314
74	476	2.55699677	.844081139
75	570	3.01004702	.993635965
76	535	2.84135808	.937950657
77	467	2.51361957	.829762059

2300

LOC.#	RATIO	CALC(M/S)
0	.9929	3.25187621
1	1.0286	3.36879834
2	1.0515	3.4437986
3	1.0214	3.34521741
4	.9714	3.18146092
5	.9242	3.0268748
6	1.0715	3.5093014
7	1	3.27512963
8	.9814	3.21421222
9	1.0071	3.29838305
10	.9571	3.13462657
11	.8399	2.75078138
12	.96426	3.1580765
13	.95	3.11137315
14	.8542	2.79761573
15	.5926	1.94084182
16	.7141	2.33877007
17	.8399	2.75078138

REFERENCE = 625
AVERAGE VELOCITY (M/S)= 3.04988441

20	.937	3.02363602
21	.9928	3.20369888
22	.927	2.9913667
23	.8382	2.70481507
24	.8563	2.76967641
25	.9599	3.09753279
26	.9313	3.00524251
27	1	3.22693279
28	.947	3.05590575
29	.811	2.61704249
30	.9714	3.13464251
31	.9599	3.09753279
32	.7838	2.52926992
33	.9284	2.9958844
34	.9027	2.91295223
35	.8382	2.70481507
36	.9571	3.08849737
37	.9728	3.13916022

REFERENCE = 615
AVERAGE VELOCITY (M/S)= 2.96103353

40	.7061	2.24450546
41	.93	2.95622444
42	.9902	3.14758434
43	.5969	1.89738749
44	.993	3.1564848
45	1.0504	3.33894424
46	.7495	2.3824626
47	.944	3.00072674
48	1.0014	3.18318618
49	.7495	2.3824626
50	1	3.17873595
51	1.0868	3.45465024
52	.6781	2.15550085
53	.944	3.00072674
54	1.0084	3.20543733
55	.5969	1.89738749
56	.93	2.95622444
57	.8796	2.79601614

REFERENCE = 605
AVERAGE VELOCITY (M/S)= 2.79636934

60	1.1018	3.10405669
61	.9539	2.68738399
62	.887	2.49890932
63	1.062	2.99192976
64	1	2.81725966
65	.9014	2.53947786
66	1.1861	3.34155168
67	1	2.81725966
68	.93	2.62005148
69	1.2243	3.449171
70	.9539	2.68738399
71	.8664	2.44087377
72	1.0396	2.9293866
73	.8759	2.46763774
74	.8441	2.37804888
75	.9936	2.7992292
76	.938	2.64258956
77	.8298	2.33776207

REFERENCE = 530
AVERAGE VELOCITY (M/S)= 2.75277572

578.265201	625
559.830211	615
525.665296	605
516.620389	530

AVERAGE OF ABOVE= 545.095274

2300 correction factor determination

LOC.#	RATIO	CALC(M/S)
0	.9929	3.08438496
1	1.0286	3.1952849
2	1.0515	3.26642239
3	1.0214	3.17291852
4	.9714	3.01759649
5	.9242	2.87097249
6	1.0715	3.3285512
7	1	3.10644069
8	.9814	3.04866609
9	1.0071	3.12849642
10	.9571	2.97317439
11	.8399	2.60909954
12	.96426	2.9954165
13	.95	2.95111866
14	.8542	2.65352164
15	.5926	1.84087675
16	.7141	2.2183093
17	.8399	2.60909954

REFERENCE = 590
AVERAGE VELOCITY (M/S)= 2.89279696

20	.937	2.95137933
21	.9928	3.12713916
22	.927	2.91988115
23	.8382	2.64017732
24	.8583	2.70348866
25	.9599	3.02351015
26	.9313	2.93342536
27	1	3.14981785
28	.947	2.9828775
29	.811	2.55450228
30	.9714	3.05973306
31	.9599	3.02351015
32	.7838	2.46882723
33	.9284	2.92429089
34	.9027	2.84334057
35	.8382	2.64017732
36	.9571	3.01469066
37	.9728	3.0641428

REFERENCE = 599
AVERAGE VELOCITY (M/S)= 2.89027286

40	.7061	2.31937539
41	.93	3.05483517
42	.9902	3.25257826
43	.5969	1.96067862
44	.993	3.26177561
45	1.0504	3.45032136
46	.7495	2.46193436
47	.944	3.10082193
48	1.0014	3.28936768
49	.7495	2.46193436
50	1	3.284769
51	1.0868	3.56988695
52	.6781	2.22740186
53	.944	3.10082193
54	1.0084	3.31236105
55	.5969	1.96067862
56	.93	3.05483517
57	.8796	2.88928281

REFERENCE = 627
AVERAGE VELOCITY (M/S)= 2.88964779

60	1.1018	3.2580562
61	.9539	2.82071139
62	.887	2.62288605
63	1.062	3.14036638
64	1	2.95703049
65	.9014	2.66546729
66	1.1861	3.50733387
67	1	2.95703049
68	.93	2.75003836
69	1.2243	3.62029243
70	.9539	2.82071139
71	.8664	2.56197122
72	1.0398	3.07472031
73	.8759	2.59006301
74	.8441	2.49602944
75	.9936	2.9381055
76	.938	2.7736946
77	.8298	2.4537439

REFERENCE = 559
AVERAGE VELOCITY (M/S)= 2.88934735

545.672327 590/625 = .944 DC
545.14862 599/616 = .974 UE
545.018929 627/605 = 1.036 DC
544.956593 559/530 = 1.055 UC

AVERAGE OF ABOVE= 545.199117

CHECKED DUCT VELOCITY PROFILE

DATE 22/5/85
DAMPER # 5
FAN SPEED 2300 RPM
AMBIENT TEMP. 14.7 Deg.C
BAROMETRIC PRESSURE 29.52 in.Hg

ALL READINGS TAKEN FROM VELOMETER IN FPM!

DOWNSTREAM EVAPORATOR

LOCATION #	FPM	LOCATION #	FPM	LOCATION #	FPM	
0	640	1	675	2	695	
3	670	4	620	5	610	
6	675	7	640	8	640	
9	650	10	615	11	550	598.9
12	615	13	605	14	550	
15	360	16	440	17	530	

UPSTREAM EVAPORATOR

LOCATION #	FPM	LOCATION #	FPM	LOCATION #	FPM	
20	650	21	650	22	620	
23	600	24	560	25	685	
26	645	27	640	28	670	627.8
29	580	30	640	31	690	
32	530	33	600	34	630	
35	580	36	650	37	680	

DOWNSTREAM CONDENSER

LOCATION #	FPM	LOCATION #	FPM	LOCATION #	FPM	
40	590	41	610	42	760	
43	500	44	625	45	790	
46	600	47	610	48	760	652.5
49	640	50	660	51	800	
52	580	53	660	54	780	
55	510	56	620	57	650	

UPSTREAM CONDENSER

LOCATION #	FPM	LOCATION #	FPM	LOCATION #	FPM	
60	620	61	540	62	580	
63	660	64	580	65	580	
66	740	67	540	68	560	590.8
69	770	70	520	71	540	
72	610	73	520	74	530	
75	600	76	525	77	530	

DUCT VELOCITY PROFILE FOR VARIOUS DAMPER POSITIONS

Located between the intake and exhaust ducting, is a damper which permits the bleeding of warm exhaust air into the cooler intake air. This damper allows some flexibility in achieving a desired intake air temperature since normally, the temperature of the duct intake air would be fixed at outdoor ambient air conditions.

The damper angle can be set anywhere from 0 to 75 degrees open (0 degrees is completely closed) by pivoting the damper into the intake air stream (see the end of this Appendix for further details). As a result , the velocity profile for fan speeds of 2300 RPM or greater downstream of the damper {upstream condenser} are found to be influenced by damper position. To account for this, a duct velocity profile at a fan speed of 2300 rpm was determined for the 5 damper open positions commonly used. This profile was normalized with respect to the 5th damper position (0 degrees open) as follows:

eg: Duct Velometer probe location #60
damper position #1.........V{1} = 600 FPM
damper position #5.........V{5} = 590 FPM

From velometer calibration;

$$C\{I\} = .262 + 4.819E\text{-}3 * V\{I\} \quad \{m/s\}$$

Therefore: C{1} = 3.1546 m/s
C{5} = 3.1064 m/s and X{1} = C{1}/C{5} = 1.016

The normalized velocity distribution determined earlier for a damper position of 0 degrees open is then corrected in the data acquisition program by multipying each of it's 18 grid velocities by X{I}, where I ranges from 0 to 5 depending on the damper position used.

DUCT VELOCITY PROFILE
FOR
VARIOUS DAMPER POSITIONS

DATE 25/7/84
FAN SPEED 2300 RPM
AMBIENT TEMPERATURE 22.6 Deg.C

ALL READINGS TAKEN FROM VELOMETER IN FPM!

UPSTREAM CONDENSER

LOCATION #	DAMPER POSITION # 1	2	3	4	5
60	600	600	600	595	590
61	525	530	530	520	510
62	585	595	595	590	580
63	685	680	670	670	655
64	595	595	595	595	580
65	640	640	635	635	610
66	760	766	755	755	765
67	580	580	585	585	565
68	585	590	590	590	575
69	760	765	765	765	750
70	545	545	545	540	530
71	565	570	570	565	550
72	595	595	595	595	580
73	530	530	530	530	495
74	550	550	550	545	535
75	635	615	615	610	600
76	570	570	570	565	545
77	535	535	530	530	520

DUCT VELOCITY PROFILE FOR VARIOUS DAMPER POSITIONS

DATE: 25/7/84
FAN SPEED: 2300

REFERENCE: DAMPER POSITION #5
UPSTREAM CONDENSER

LOCATION #	DAMPER POSITION RATIO			
	1	2	3	4
60	1.016	1.016	1.016	1.008
61	1.027	1.035	1.035	1.018
62	1.008	1.024	1.024	1.016
63	1.042	1.035	1.021	1.021
64	1.024	1.024	1.024	1.024
65	1.045	1.045	1.038	1.038
66	.994	.994	.988	.988
67	1.024	1.024	1.032	1.032
68	1.016	1.024	1.024	1.024
69	1.012	1.019	1.019	1.019
70	1.026	1.026	1.026	1.017
71	1.025	1.033	1.033	1.025
72	1.024	1.024	1.024	1.024
73	1.064	1.064	1.064	1.064
74	1.025	1.025	1.025	1.017
75	1.053	1.023	1.023	1.015
76	1.042	1.042	1.042	1.033
77	1.026	1.026	1.017	1.017

DUCT STATIC PRESSURE MEASUREMENTS

Duct static pressure measurements were taken at each of the four measurement stations {UC,DC,UE,DE} using the Lambrecht type 655 manometer for all 3 fan speeds and 5 damper positions. The manometer contained fluid of specific gravity .827 and was positioned to read 1/2 scale. The following data sheet shows the amount of manometer fluid in mm and the conversion to mm of Hg using the equation;

$$P = (SG)(DW)(SF)(g)(L)(CF)$$

$$P(\text{mm of Hg}) = (.827)(997.1)(.5)(9.81)(L)/1.333E5$$

Where:

- SG : Specific gravity of manometer fluid
- DW : Density of water
- SF : scale factor used with manometer
- g : local gravity
- L : length along the manometer in mm of manometer fluid
- CF : Units conversion factor

These static pressure measurements were incorporated into the computer data acquisition program for later use.

DUCT STATIC PRESSURE CALIBRATION

DATE 20/7/84
BAROMETRIC PRESSURE 29.452
MANO. LIQ.SPECIFIC GRAVITY .827
SCALE 1/2

ALL MANOMETER READINGS IN MM

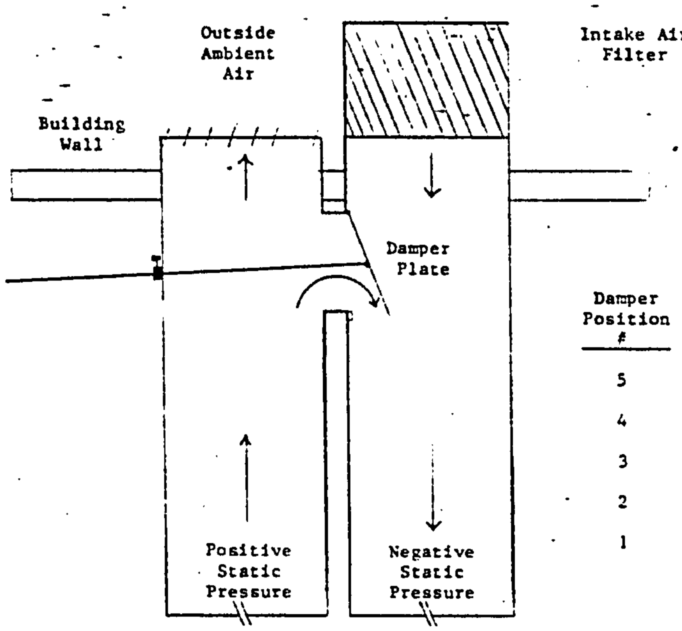
DAMPER #	FAN SPEED 1150 mm	1150 mm.Hg	1700 mm	1700 mm.Hg	2300 mm	2300 mm.Hg
5	UC= -25.0 DC= -42.0 UE= 26.0 DE= 9.5	-.758 -1.274 .789 .288	UC= -52.0 DC= -91.5 UE= 49.5 DE= 13.5	-1.578 -2.776 1.502 .410	UC= -91.5 DC= -166.0 UE= 87.0 DE= 18.0	-2.776 -5.036 2.639 .546
4	UC= -21.5 DC= -41.5 UE= 25.5 DE= 6.5	-.652 -1.259 .774 .197	UC= -43.0 DC= -88.0 UE= 50.0 DE= 10.5	-1.305 -2.670 1.517 .319	UC= -82.0 DC= -161.0 UE= 84.0 DE= 13.5	-2.488 -4.884 2.548 .410
3	UC= -20.5 DC= -41.5 UE= 25.5 DE= 6.5	-.622 -1.259 .774 .197	UC= -44.0 DC= -88.0 UE= 49.0 DE= 8.5	-1.335 -2.670 1.487 .258	UC= -79.5 DC= -159.0 UE= 84.0 DE= 11.0	-2.412 -4.824 2.548 .334
2	UC= -20.0 DC= -41.0 UE= 26.0 DE= 6.5	-.607 -1.244 .789 .197	UC= -43.5 DC= -88.0 UE= 49.0 DE= 9.0	-1.320 -2.670 1.487 .273	UC= -80.5 DC= -160.0 UE= 84.0 DE= 10.5	-2.442 -4.854 2.548 .319
1	UC= -20.5 DC= -42.0 UE= 25.5 DE= 6.5	-.622 -1.274 .774 .197	UC= -45.0 DC= -89.0 UE= 48.5 DE= 7.5	-1.365 -2.700 1.471 .228	UC= -82.5 DC= -160.0 UE= 84.0 DE= 11.5	-2.503 -4.854 2.548 .349

LOCATION	PROBE POSITION # USED
UE	27
DE	7
UC	67
DC	50

DAMPER POSITION DATA

The mixing chamber, which was used to direct the warm duct exhaust air into the cold duct intake air supply, was opened or closed using a hinged, flat plate damper located in the cold air supply duct over the chamber opening. Unfortunately when the damper plate was in the open chamber position, as seen in Figure A.6 below, the plate protruded into the cold air supply stream and thus, changed the static pressure and velocity profiles of the air flow downstream of the chamber.

To allow for the use of the mixing chamber in this study, both velocity and static pressure profiles were taken for the different damper positions and used in the data aquisition program.



Damper Position #	Angle of Damper Opening (Degrees)	% Damper Open
5	0	0
4	15	26
3	30	40
2	50	67
1	75	100

FIGURE A.6
SCHEMATIC OF MIXING CHAMBER/DAMPER
ARRANGEMENT ON TSHE TEST APPARATUS

When the need arose for a substantially warmer intake duct air temperature, quite often the mixing chamber could not divert an adequate amount of air from the warm exhaust air duct into the cold supply air duct to achieve the temperature desired. This created difficulty when a minimum supply air temperature of 25 degrees C. was needed to keep the loop pressures above that of atmospheric air. Figure A.7 shows the damper arrangement inside the ducting looking up into the supply air side of the system.

To reduce the dampers impact on the flow profiles and increase the amount of flow between the two ducts when needed, a larger chamber opening with a sliding plate damper is suggested.

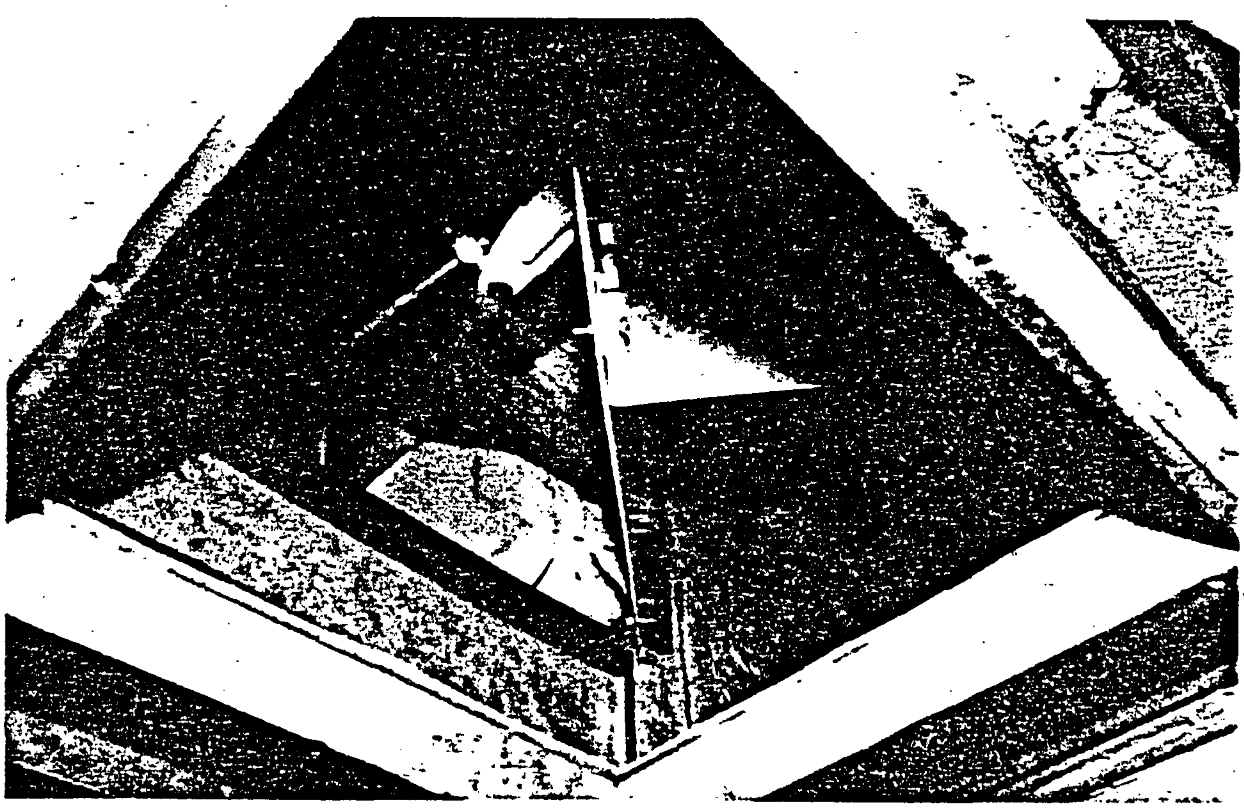


FIGURE A.7
DAMPER CONFIGURATION INSIDE THE SUPPLY AIR DUCT

APPENDIX B

R-11 FLOWMETER CALIBRATIONS

FLOWMETERS:

Eight, size 3, Gilmont taperred tube and float flowmeters were purchased and installed into the TSHE test apparatus to measure the refrigerant volume flow rate present in each of the 8 condensate return lines, and to provide a visual indication of the sytems dynamic flow stability. The flowmeters were placed into the TSHE loops immediately upstream from the evaporator and as such, gave an indication of the R-11 evaporation rate in each of the 8 separate rows of evaporator tubes.

Before installation began, the calibration of the flowmeters were checked to ensure that the volume flow rate and pressure drop versus float scale readings would be within the expected limits suggested by the manufacturer.

The testing apparatus shown in Figure B.1 was constructed and placed inside the large walkin cooler which was previously located in room 203 Essex Hall. A cooler temperature of less than 10 degrees C was used to ensure that the calibration would be conducted with subcooled R-11. The following testing procedure was used to calibrate each of the eight flowmeters:

1. The high elevation reservoir was filled with liquid R-11 which had been cooled to below 10 degrees C.
2. The valve located immediately after the flowmeter was opened and the system primed.
3. Once the liquid R-11 began to flow, the valve was adjusted in order to achieve the desired float level in the flowmeter.
4. A stop watch was then used to find the amount of time needed to collect 0.7 lbs of liquid R-11 in the bottle

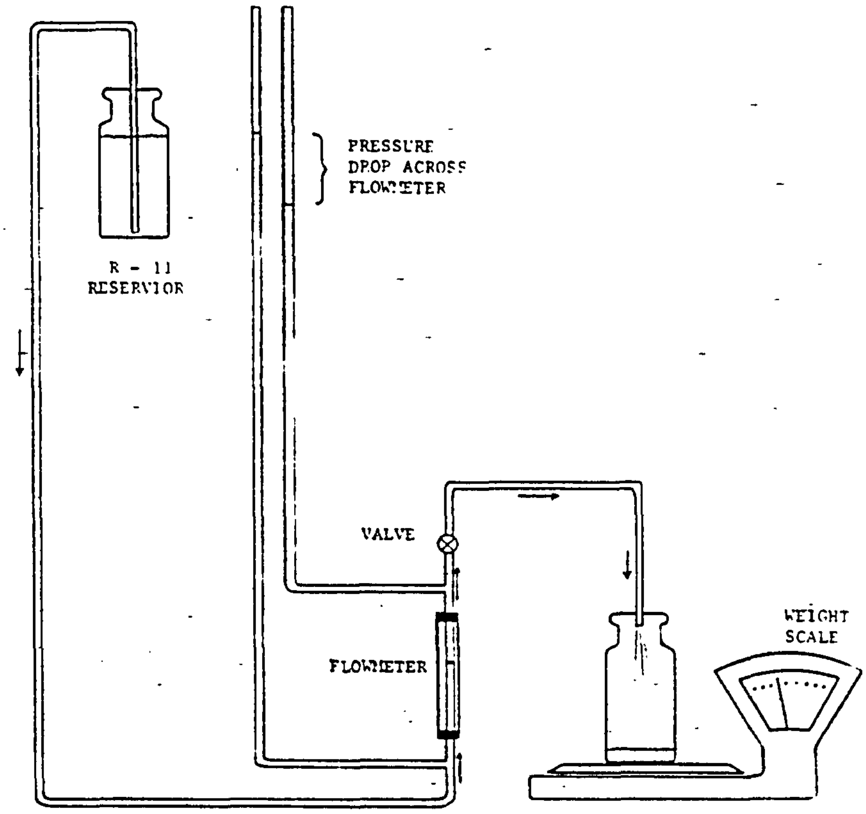


FIGURE B.1
FLOWMETER CALIBRATION APPARATUS

located on the weighing scale. This time was recorded.

5. The difference in liquid R-11 height between the two presssure tubes was simultaneously measured and recorded.

6. Steps 1 through 4 were then repeated as many times as necessary so that a good indication of the flow rate versus scale reading could be determined.

From this experiment, the flowrate of R-11 {ml/min} and the pressure drop across the flowmeter {Pa} was calculated using the density of R-11 at the coldroom temperature and the following equation:

$$Q = \frac{(0.7/t)(1.699011E6)}{\rho}$$

where: Q is the flow rate in ml/min

t is the time needed to collect 0.7 lbs of R-11 in seconds

ρ is the density of R-11 in lbs/ft3

The pressure drop was also calculated using the equation:

$$P = \rho * h * g$$

$$= \rho * h * (9.81E-3)$$

where: ρ : density of R-11 (Kg/m3)

h : the difference in height of R-11 (mm)

P : the pressure drop (Pa)

When a comparison was made between the coldroom calibrated flow rates and those suggested by the manufacturer, it was found that the method of calculation supplied by the Gilmont company seriously overpredicted the actual flow rate found for the liquid refrigerant R-11 used.

This created concern since the flowrate capacity measured by the flowmeters was much less than the flowrates expected through the individual loops by previous calculations. To ensure the usefulness of the flowmeters, a second float was designed prior to installation to operate at volume flow rates greater than the range covered by the original manufacture's supplied float. With some experimentation, a nail shaped float made of brass was found to meet this requirement. Eight separate brass floats were then machined and placed under the original black glass float supplied with each of the flowmeters. Each flowmeter was then calibrated with this arrangement by the procedure previously explained. A summary of the calibration curvefits determined for the two float arrangement can be seen in Table B.1 .

FLOWMETER #	GLASS OR BRASS FLOAT	FLOW RATE =B0 + B1 * SCALE READING (ML/MIN) B0	(MM) B1
0	G	3.53034	1.4064
	B	134.835	5.7521
1	G	5.5607	1.3709
	B	147.156	5.7376
2	G	2.7222	1.3988
	B	219.86	5.8469
3	G	7.2705	1.3488
	B	152.354	5.9169
4	G	2.3381	1.3813
	B	107.952	5.8637
5	G	2.3182	1.3684
	B	131.201	5.8506
6	G	1.833	1.3674
	B	77.662	5.936
7	G	4.5142	1.3579
	B	105.741	5.8039

TABLE B.1
FLOWMETER CALIBRATION CURVEFITS

FLOWMETER FLOWRATE CORRECTION:

The Gilmont flowmeter essentially consists of an internally tappered glass tube which houses a symetrical ball or 'float'. The float's position within the taperred tube is influenced by a variety of factors, one of which is the density of the fluid used. Since the temperature of the refrigerant {and thus it's density} cycled in the TSHE system will not always be at the same temperature as when the flowmeters were calibrated in the coldroom, a correction to the density of the R-11 at the temperature found in the TSHE system must be made. The following equations supplied by Gilmont were used to calculate the density correction and were included in the TSHE data acquisition program:

$$Q\{liq.T\} = K\{liq\} * Q\{liq.C\} \qquad \text{Eqn. B.1}$$

where: $$K\{liq\} = \frac{(DF-DL.T)}{(DF-DL.C)} * \frac{DL.C}{DL.T}$$

Q{liq.T} : Volume flow rate of liquid R-11 at temp. T in the TSHE system {ml/min}

Q{liq.C} : Volume flow rate of liquid R-11 at calibration temp. {ml/min}

DF : Density of float material {kg/m3}

DL.C : Density of R-11 at calibration temp.{Kg/m3}

DL.T : Density of R-11 at temp. T in TSHE system{kg/m3}

DF{brass} = 8530 Kg/m3

DF{glass} = 2530 Kg/m3

The actual calibration data obtained and a copy of the supplied manufacturer`s calibration method can be found in the remainder of this Appendix.

R-11 FLOW METER CALIBRATION SHEET

FLOW METER NUMBER F θ FLOAT O
DATE 18/5/84

Ambient Temperature: 10 Deg.C, 50 Deg.F
Ambient Pressure: 29.38 in.Hg.
R-11 Density: 94.347 Lbs/ft3, 1511.3 Kg/m3

All flow meter readings taken at top of float ball and at top of nail shaped float.

FLOW METER READING (mm) GLASS FLOAT	BRASS FLOAT	TIME NEEDED TO PASS .7 Lbs R-11 (s)	FLOW RATE CALCULATED (ml/min)
20		384.91	32.75
50		171.73	73.40
80		111.75	112.80
100	2	85.89	146.77
	18	53.80	234.31
	50	29.28	430.52
	79	21.59	583.87
	100	17.73	710.98

R-11 FLOW METER PRESSURE DROP DATA

FLOW METER NUMBER F θ
DATE: 24/5/84

Ambient temperature: 10 Deg.C
R-11 Density: 1510 Kg/m3

FLOW METER READING (mm) GLASS	BRASS	PRESSURE DIFFERENCE (mm R-11)	PRESSURE DROP CALCULATED ACROSS FLOW METER (Pa)
20		5	74.1
50		25	370.3
75		30	444.4
100		30	444.4
	20	40	592.5
	50	70	1036.9
	75	115	1703.5
	100	180	2666.4

****

#	X	Y	CALC.Y
1	20	32.75	31.6582314
2	50	73.4	73.8500681
3	80	112.8	116.041905
4	100	146.77	144.169796

Y=BO +B1X
WHERE BO=3.53034023
B1=1.40639456

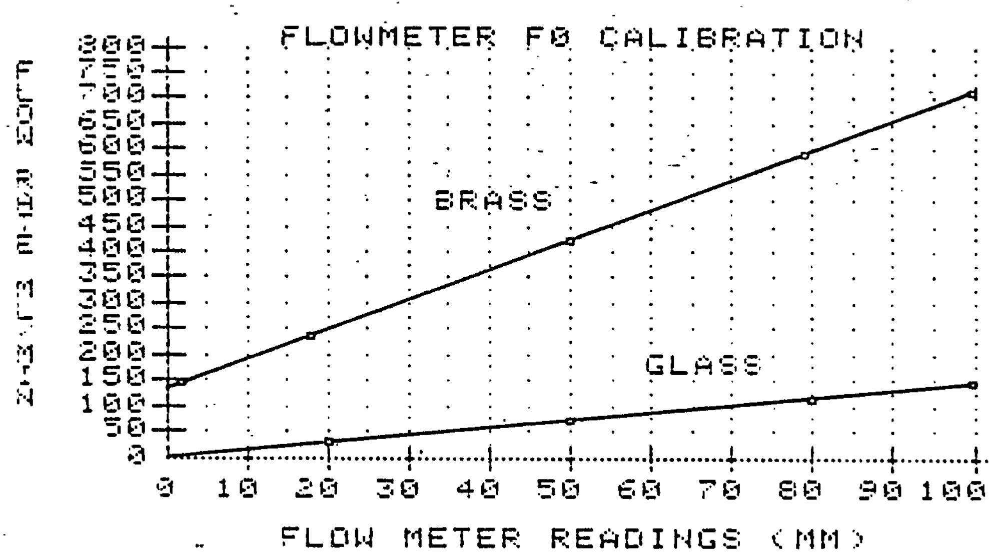
CORRELATION OF FIT.R=.998718544
(R=1 IS A PERFECT FIT)

****

#	X	Y	CALC.Y
1	2	146.77	146.339366
2	18	234.31	238.373051
3	50	430.52	422.440422
4	79	583.87	589.251476
5	100	710.98	710.045688

Y=BO +B1X
WHERE BO=134.835155
B1=5.75210533

CORRELATION OF FIT.R=.999746734
(R=1 IS A PERFECT FIT)



F0

GLASS FLOAT CALIBRATION

USING ONLY 3 4 POINTS

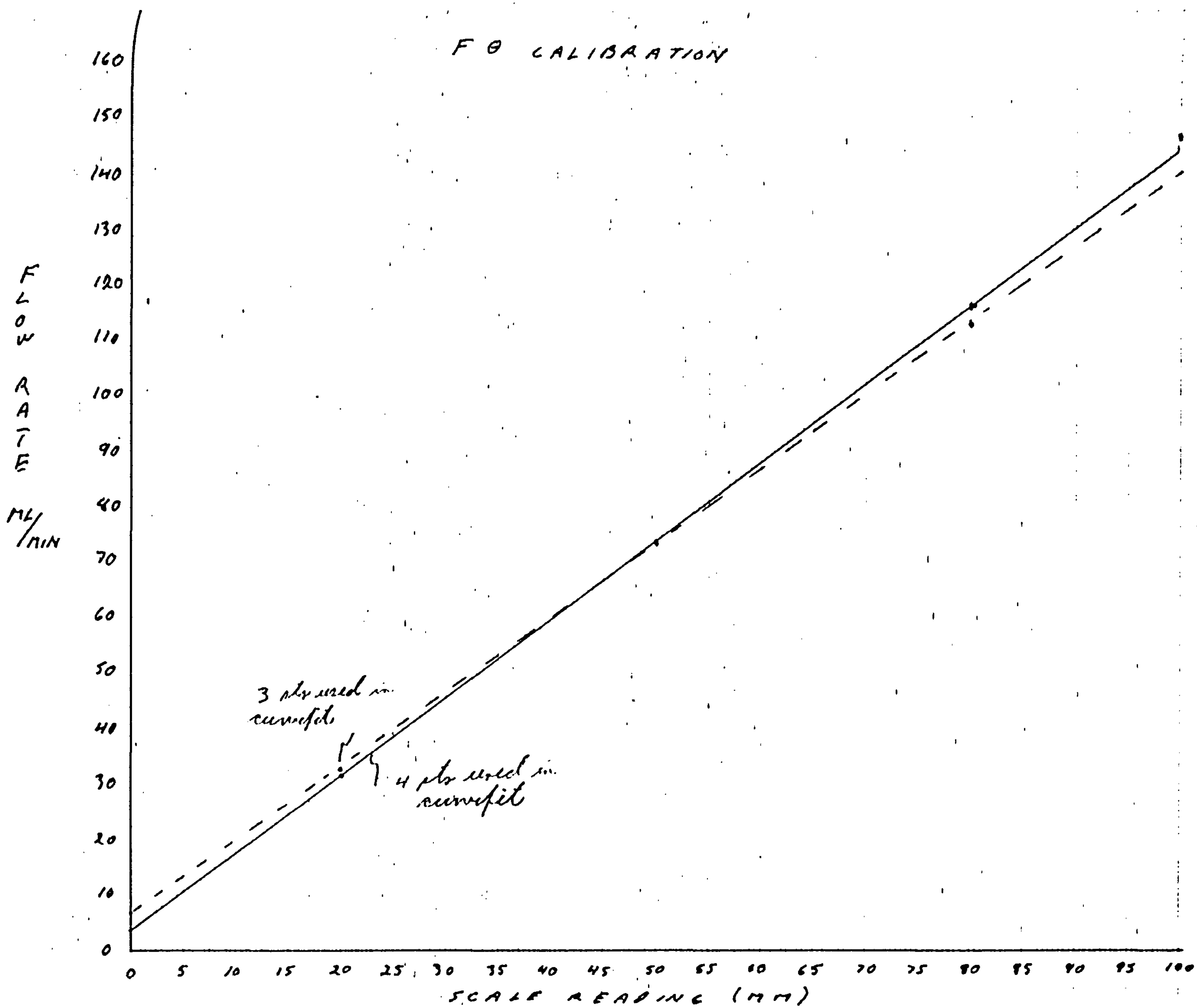
****

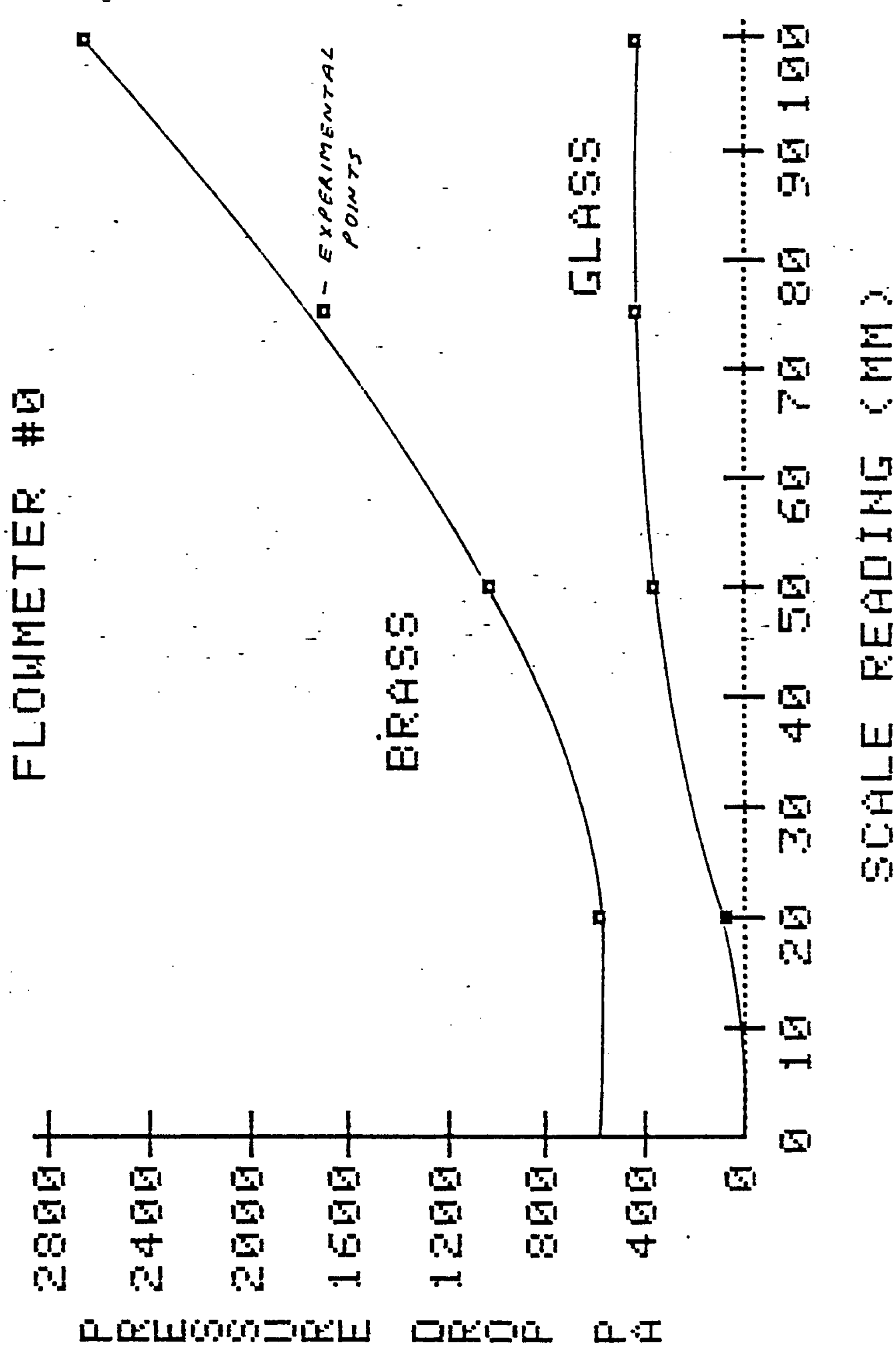
#	X	Y	CALC.Y
1	20	32.75	32.9583334
2	50	73.4	72.9833333
3	80	112.8	113.008333

Y=B0 +B1X
WHERE B0=6.27500008
B1=1.33416666 — too low from average. ∴ use 4 pts in curvefit

CORRELATION OF FIT.R=.999959359
(R=1 IS A PERFECT FIT)

* see curvefit of flowmeter #F1 for averages


F θ CALIBRATION
FLOW RATE ML/MIN
160
150
140
130
120
110
100
90
80
70
60
50
40
30
20
10
0
0
5
10
15
20
25
30
35
40
45
50
55
60
65
70
75
80
85
90
95
100
SCALE READING (MM)
3 pts used in curvefit
4 pts used in curvefit


FLOWMETER #0
BRASS
GLASS
- EXPERIMENTAL POINTS
SCALE READING (MM)
0 10 20 30 40 50 60 70 80 90 100
0 400 800 1200 1600 2000 2400 2800

R-11 FLOW METER CALIBRATION SHEET

FLOW METER NUMBER F1 FLOAT #1
DATE 5/23/84

Ambient Temperature: 5 Deg.C, 41 Deg.F
Ambient Pressure: 29.31 in.Hg.
R-11 Density: 95.068 Lbs/ft3, 1522.8 Kg/m3

All flow meter readings taken at top of float ball!

FLOW METER READING (mm) GLASS FLOAT	BRASS FLOAT	TIME NEEDED TO PASS .7 Lbs R-11 (s)	FLOW RATE CALCULATED (ml/min)
20		369.15	33.89
50		172.19	72.65
75		128.66	97.23
100		87.36	143.20
	10	60.37	207.22
	50	29.40	425.51
	75	21.39	584.86
	100	17.39	719.38

R-11 FLOW METER PRESSURE DROP DATA

FLOW METER NUMBER F1
DATE: 24/5/84

Ambient temperature: 10 Deg.C
R-11 Density: 1510 Kg/m3

FLOW METER READING (mm) GLASS	FLOW METER READING (mm) BRASS	PRESSURE DIFFERENCE (mm R-11)	PRESSURE DROP CALCULATED ACROSS FLOW METER (Pa)
20		5	74.1
50		15	222.2
75		25	370.3
100		30	444.4
	20	45	666.59
	50	90	1333.2
	75	140	2073.8
	100	200	2962.6

****

#	X	Y	CALC.Y
1	20	33.89	32.9792857
2	50	72.65	74.1071428
3	100	143.2	142.653571

Y=BO +B1X
WHERE BO=5.56071425
B1=1.37092857

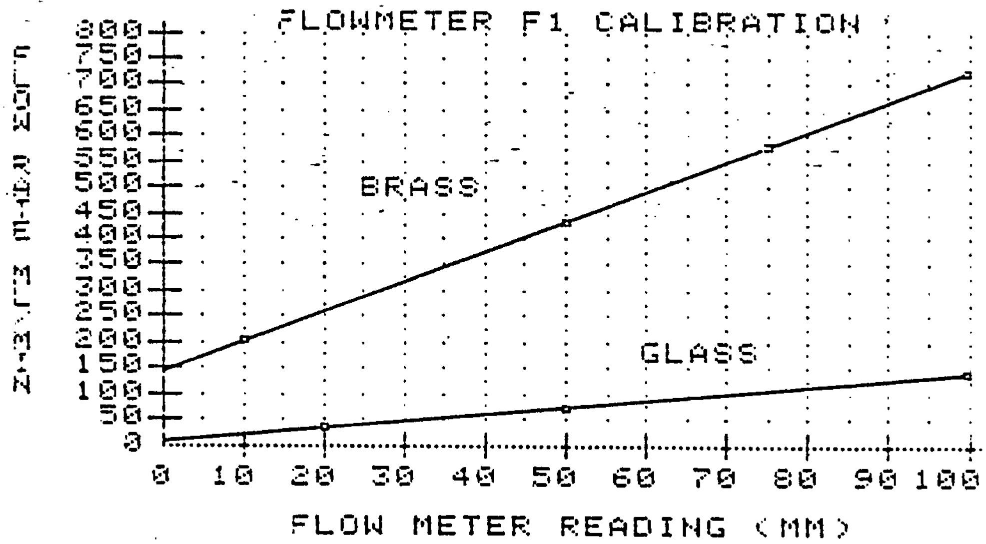
CORRELATION OF FIT:R=.999735322
(R=1 IS A PERFECT FIT)

****

#	X	Y	CALC.Y
1	10	207.22	204.532376
2	50	425.51	434.038118
3	75	584.86	577.479207
4	100	719.38	720.920297

Y=BO +B1X
WHERE BO=147.15594
B1=5.73764356

CORRELATION OF FIT.R=.999530113
(R=1 IS A PERFECT FIT)



****

#	X	Y	CALC.Y
1	20	33.89	31.9890231
2	50	72.65	71.8097336
3	75	97.23	104.993659
4	100	143.2	138.177584

Y=BO +B1X
WHERE BO=5.44188286
B1=1.32735701

CORRELATION OF FIT.R=.992833851
(R=1 IS A PERFECT FIT)

****

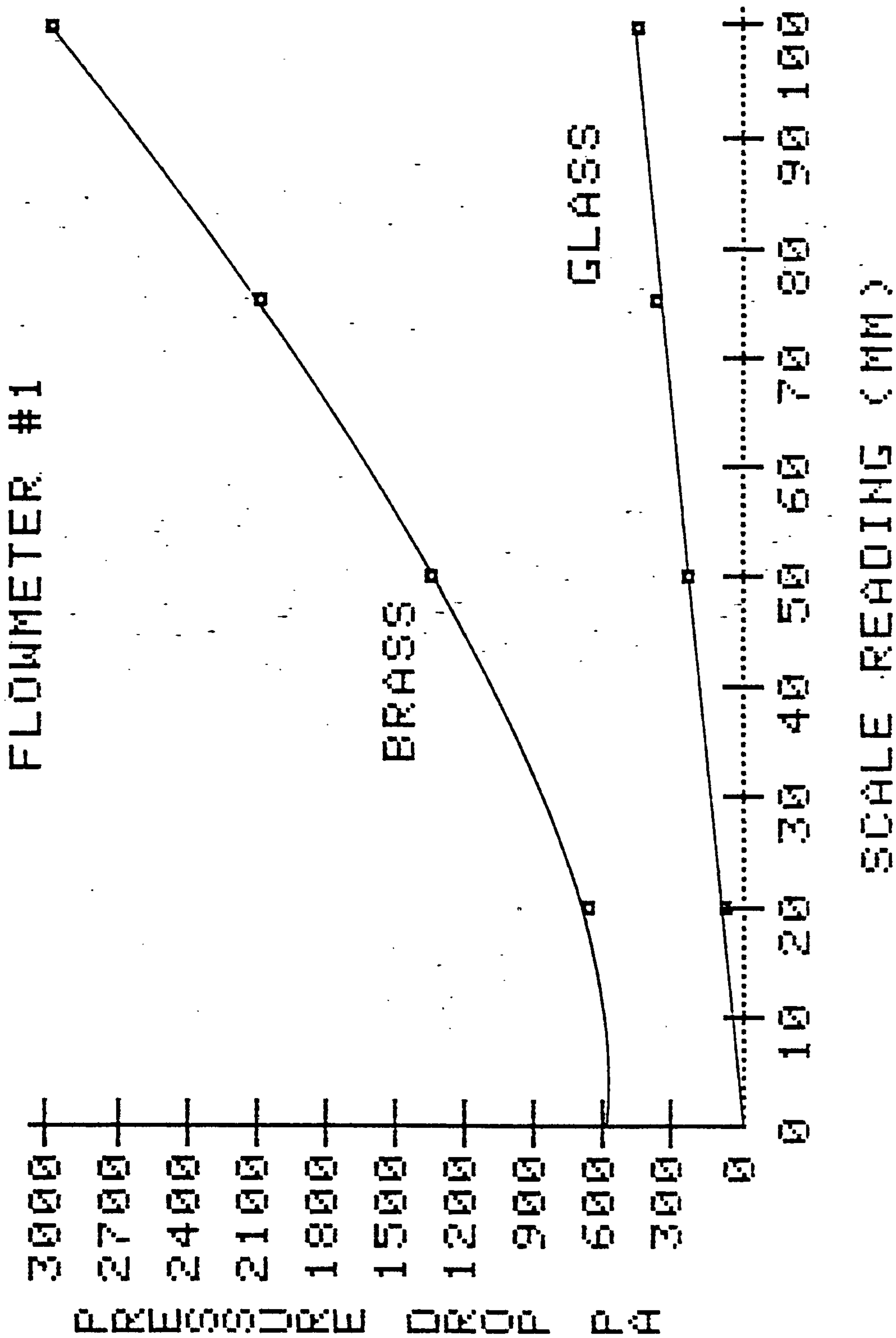
#	X	Y	CALE.Y
1	10	207.22	204.532376
2	50	425.51	434.038118
3	75	584.86	577.479207
4	100	719.38	720.920297

Y=BO +B1X
WHERE BO=147.15594
B1=5.73764356

CORRELATION OF FIT.R=.999530113
(R=1 IS A PERFECT FIT)

	slope of Glass float curve
F0	~~1.349~~ 1.406
F2	1.399
F3	1.349
F4	1.381
F5	1.368
F6	1.367
F7	1.358
ave	1.375

∴ choose slope of 3 std Glass ie neglect point # 3 from curvefitting technique


FLOWMETER #1
BRASS
GLASS
SCALE READING (MM)
0 10 20 30 40 50 60 70 80 90 100
PRESSURE DROP PA
0 300 600 900 1200 1500 1800 2100 2400 2700 3000

R-11 FLOW METER CALIBRATION SHEET

FLOW METER NUMBER F2 FLOAT #2
DATE 23/5/84

Ambient Temperature: 8 Deg.C, 46.4 Deg.F
Ambient Pressure: 29.32 in.Hg.
R-11 Density: 94.600 Lbs/ft3, 1515.3 Kg/m3

All flow meter readings taken at top of float ball!

FLOW METER READING (mm) GLASS FLOAT	FLOW METER READING (mm) BRASS FLOAT	TIME NEEDED TO PASS .7 Lbs R-11 (s)	FLOW RATE CALCULATED (ml/min)
20		395.62	31.78
50		173.25	72.57
75		120.47	104.36
100		86.77	144.89
	20	37.70	333.47
	50	24.28	517.79
	75	19.11	657.87
	100	15.66	802.81

R-11 FLOW METER PRESSURE DROP DATA

FLOW METER NUMBER F2
DATE: 24/5/84

Ambient temperature: 10 Deg.C
R-11 Density: 1510 Kg/m3

FLOW METER READING (mm) GLASS	BRASS	PRESSURE DIFFERENCE (mm R-11)	PRESSURE DROP CALCULATED ACROSS FLOW METER (Pa)
20		5	74.1
50		15	222.2
75		25	370.3
100		30	444.4
	20	60	888.8
	50	110	1629.4
	75	170	2518.2
	100	220	3258.9

****

#	X	Y	CALC.Y
1	20	31.78	30.6986501
2	50	72.57	72.6632682
3	75	104.36	107.633783
4	100	144.89	142.604298

Y=BO +B1X
WHERE BO=2.72223803
B1=1.3988206

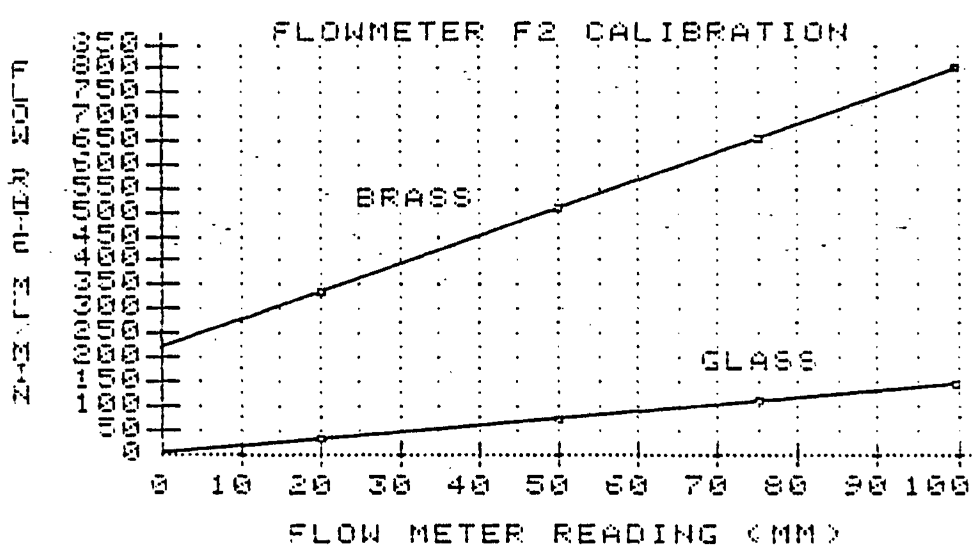
CORRELATION OF FIT.R=.998759044
(R=1 IS A PERFECT FIT)

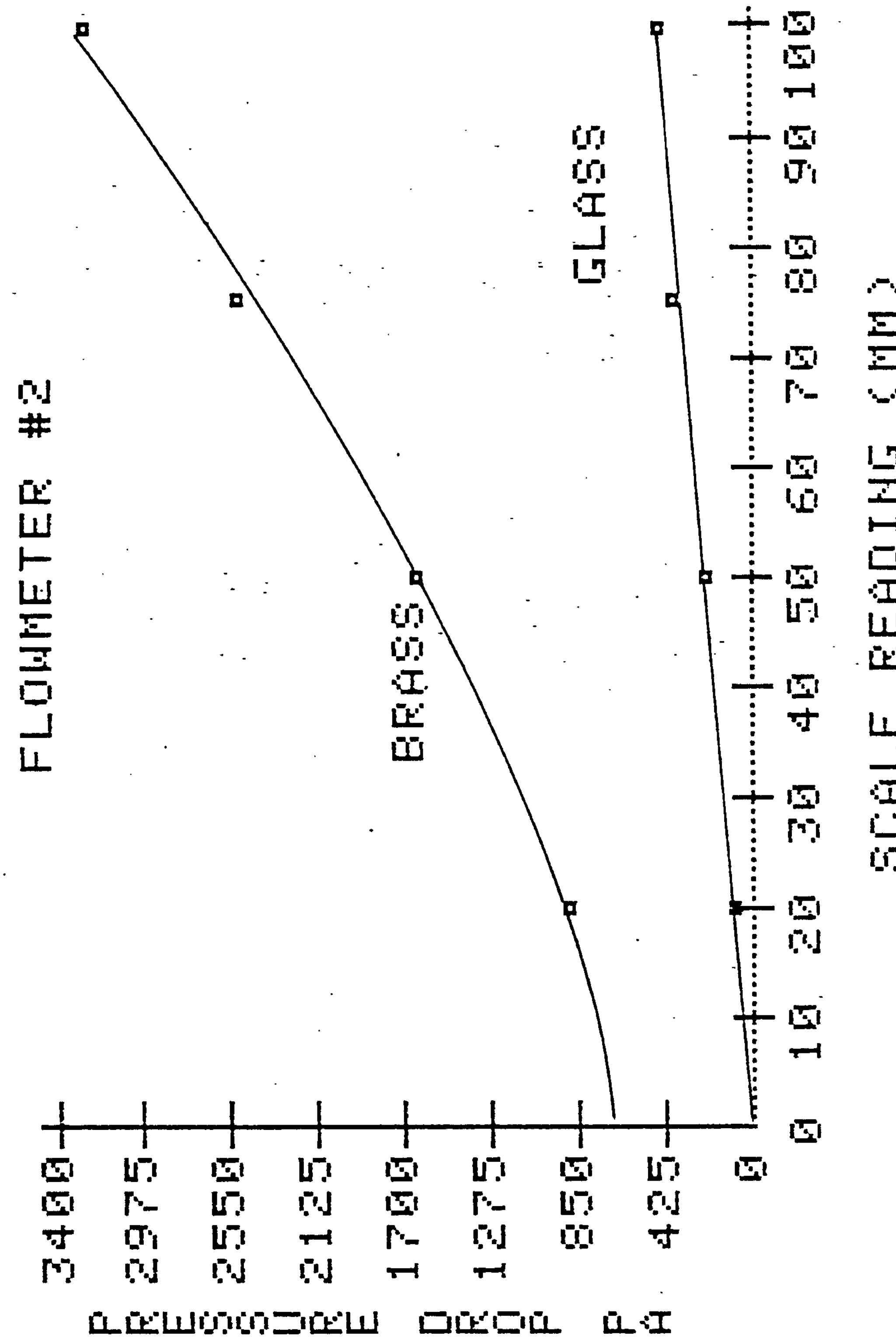
****

#	X	Y	CALC.Y
1	20	333.47	336.79984
2	50	517.79	512.207229
3	75	657.87	658.380053
4	100	802.81	804.552877

Y=BO +B1X
WHERE BO=219.861581
B1=5.84691296

CORRELATION OF FIT.R=.999810699
(R=1 IS A PERFECT FIT)




FLOWMETER #2
BRASS
GLASS
SCALE READING (MM)
PRESSURE DROP PA
3400
2975
2550
2125
1700
1275
850
425
0
0 10 20 30 40 50 60 70 80 90 100

R-11 FLOW METER CALIBRATION SHEET

FLOW METER NUMBER F3 FLOAT #3
DATE 24/5/84

Ambient Temperature: 6 Deg.C, -42.8 Deg.F
Ambient Pressure: 29.48 in.Hg.
R-11 Density: 94.778 Lbs/ft3, 1518.2 Kg/m3

All flow meter readings taken at top of float ball!

FLOW METER READING (mm) GLASS FLOAT	BRASS FLOAT	TIME NEEDED TO PASS .7 Lbs R-11 (s)	FLOW RATE CALCULATED (ml/min)
20		371.15	33.81
50		166.51	75.36
75		115.61	108.54
100		88.47	141.84
	20	46.58	269.39
	50	27.67	453.50
	75	21.28	589.68
	100	16.81	746.48

R-11 FLOW METER PRESSURE DROP DATA

FLOW METER NUMBER F3
DATE: 24/5/84

Ambient temperature: 8 Deg.C
R-11 Density: 1518.2 Kg/m3

FLOW METER READING (mm) GLASS	BRASS	PRESSURE DIFFERENCE (mm R-11)	PRESSURE DROP CALCULATED ACROSS FLOW METER (Pa)
20		5	74.5
50		10	148.9
75		24	357.4
100		30	446.8
	20	50	744.7
	50	95	1414.9
	75	150	2234.0
	100	210	3127.6

****

#	X	Y	CALC.Y
1	20	33.81	34.2474778
2	50	75.36	74.7129484
3	75	108.54	108.434174
4	100	141.84	142.1554

Y=BO +B1X
WHERE BO=7.2704973
B1=1.34884902

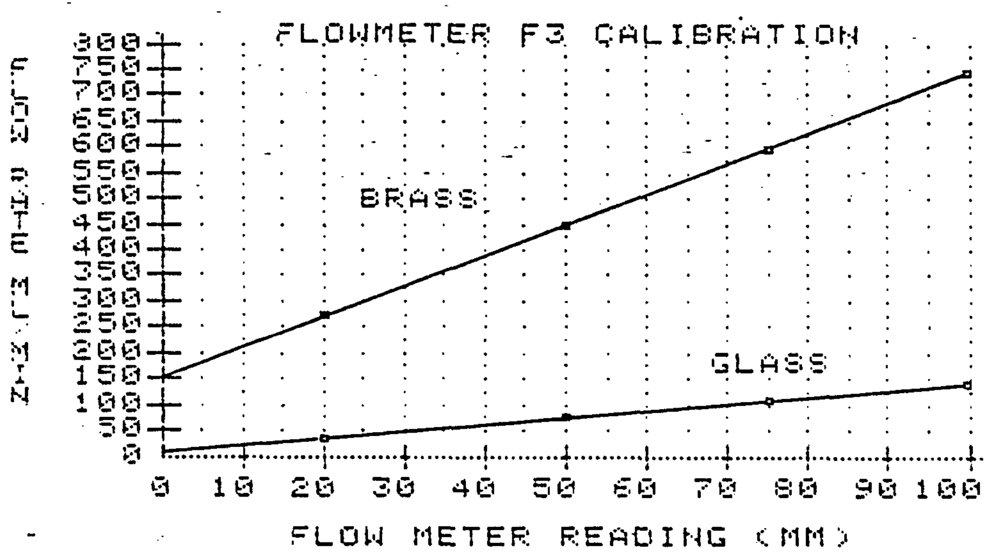
CORRELATION OF FIT.R=.999943708
(R=1 IS A PERFECT FIT)

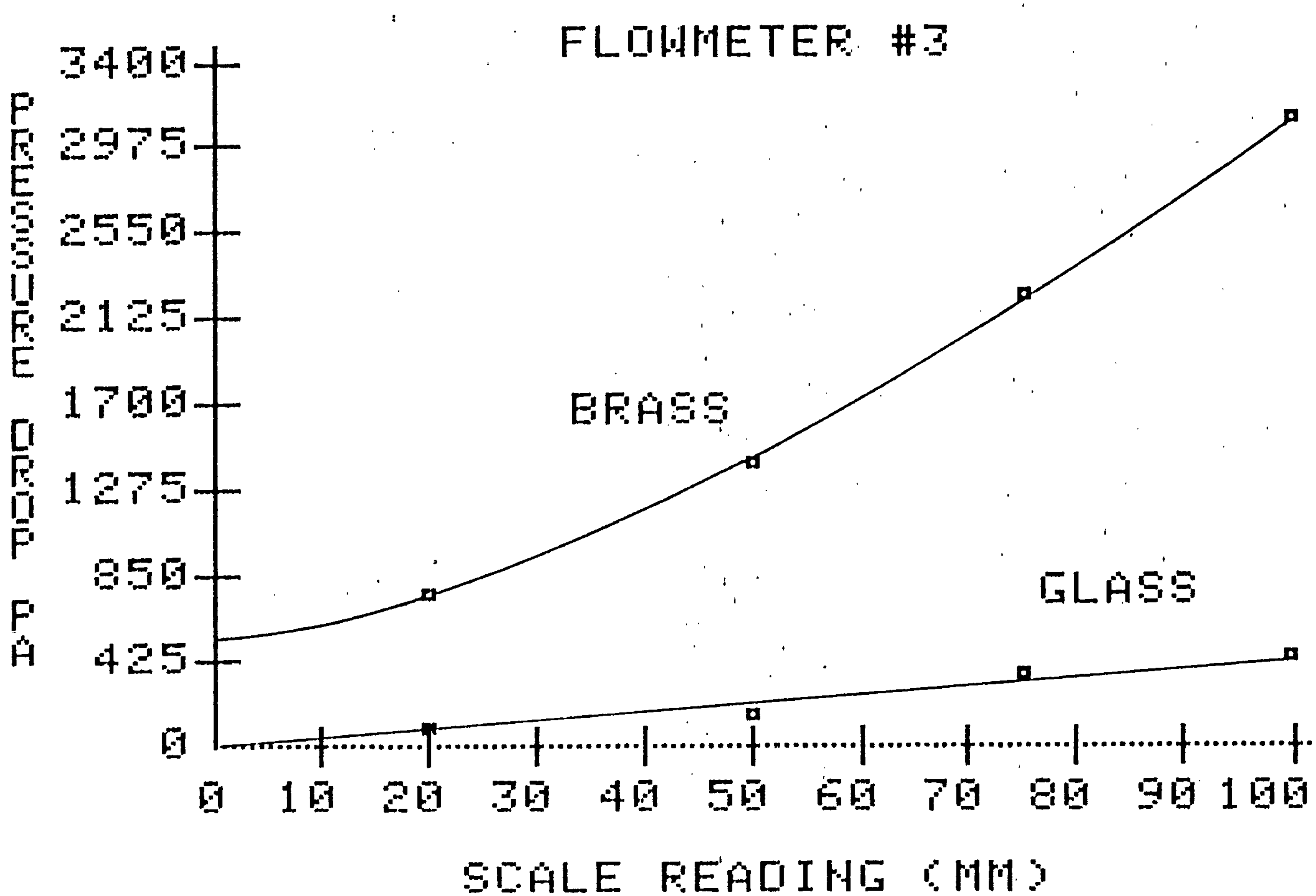
****

#	X	Y	CALC.Y
1	20	269.39	270.692486
2	50	453.5	448.199769
3	75	589.69	596.122505
4	100	746.48	744.04524

Y=BO +B1X
WHERE BO=152.354297
B1=5.91690943

CORRELATION OF FIT.R=.999687235
(R=1 IS A PERFECT FIT)




FLOWMETER #3
BRASS
GLASS
PRESSURE DROP PA
3400
2975
2550
2125
1700
1275
850
425
0
0 10 20 30 40 50 60 70 80 90 100
SCALE READING (MM)

R-11 FLOW METER CALIBRATION SHEET

FLOW METER NUMBER F4 FLOAT #4
DATE 24/5/84

Ambient Temperature: 9 Deg.C, 48.2 Deg.F
Ambient Pressure: 29.43 in.Hg.
R-11 Density: 94.508 Lbs/ft3, 1513.9 Kg/m3

All flow meter readings taken at top of float ball!

FLOW METER READING (mm) GLASS FLOAT	FLOW METER READING (mm) BRASS FLOAT	TIME NEEDED TO PASS .7 Lbs R-11 (s)	FLOW RATE CALCULATED (ml/min)
20		393.55	31.98
50		178.42	70.53
75		124.31	101.23
100	5	87.37	144.03
	20	57.44	219.08
	50	31.51	399.37
	75	23.08	545.24
	100	18.03	697.96

R-11 FLOW METER PRESSURE DROP DATA

FLOW METER NUMBER F 4
DATE: 24/5/84

Ambient temperature: 9 Deg.C
R-11 Density: 1512 Kg/m3

FLOW METER READING (mm)		PRESSURE DIFFERENCE (mm R-11)	PRESSURE DROP CALCULATED ACROSS FLOW METER (Pa)
GLASS	BRASS		
20		5	74.2
50		23	341.2
75		28	415.3
100	5	28	415.3
	20	40	593.3
	50	75	1112.5
	75	120	1779.9
	100	190	2818.2

****

#	X	Y	CALC.Y
1	20	31.98	29.9640143
2	50	70.53	71.402913
3	75	101.23	105.935329
4	100	144.03	140.467744

Y=BO +B1X
WHERE BO=2.33808184
B1=1.38129662

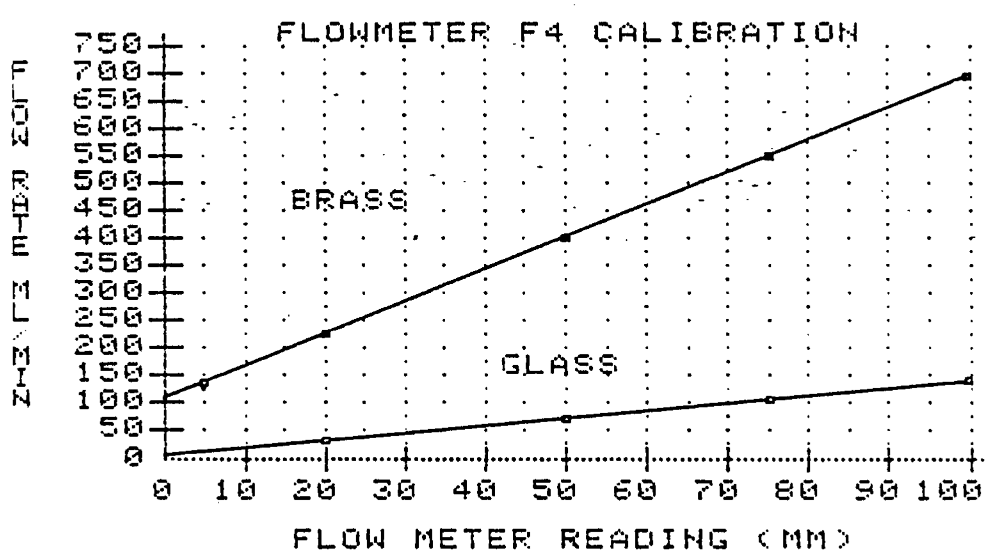
CORRELATION OF FIT.R=.997059655
(R=1 IS A PERFECT FIT)

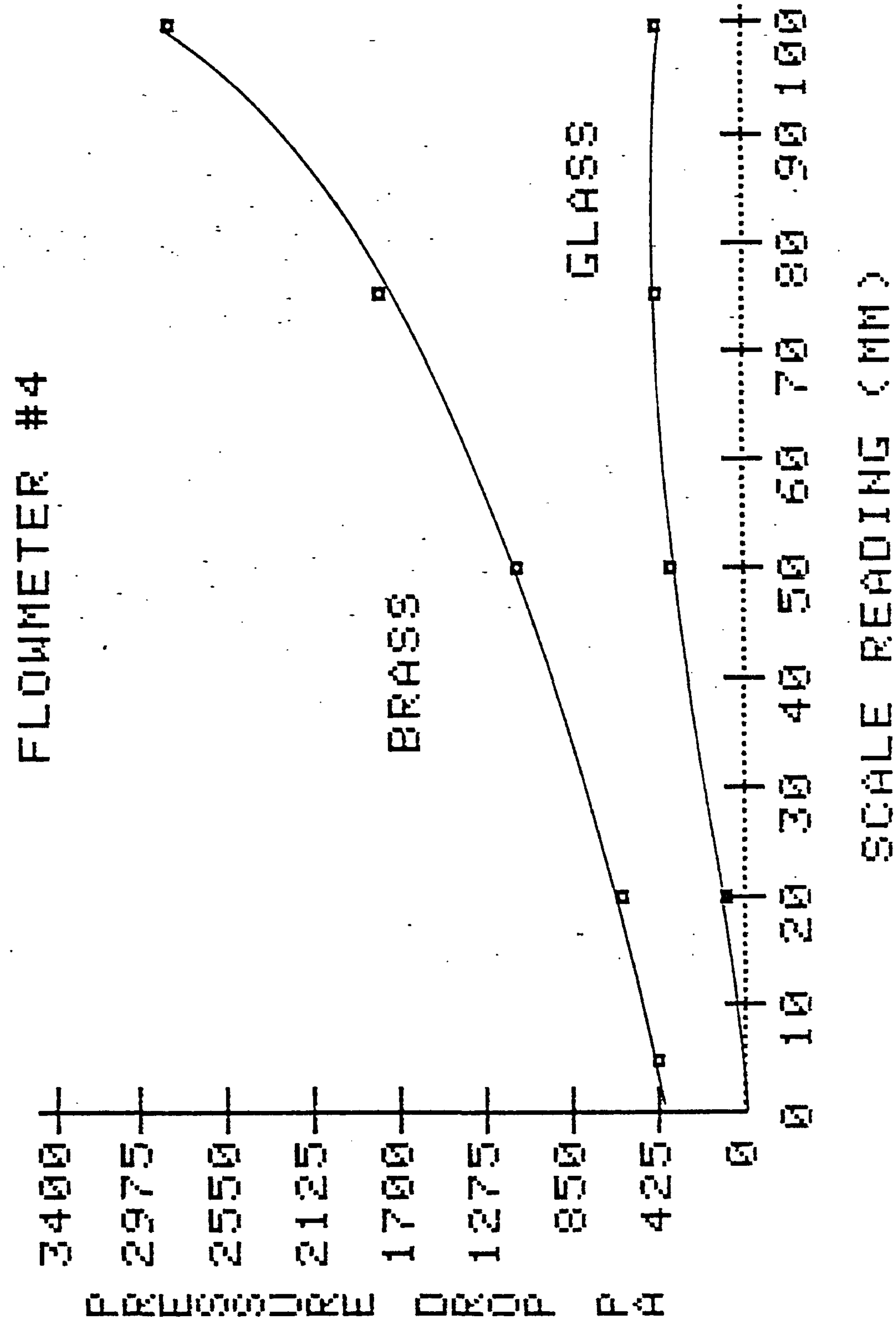
****

#	X	Y	CALC.Y
1	5	144.03	137.270504
2	20	219.08	225.22567
3	50	399.37	401.136
4	75	545.24	547.727942
5	100	697.96	694.319884

Y=BO +B1X
WHERE BO=107.952116
B1=5.86367768

CORRELATION OF FIT.R=.999745258
(R=1 IS A PERFECT FIT)




FLOWMETER #4
BRASS
GLASS
PRESSURE DROP PA
0
425
850
1275
1700
2125
2550
2975
3400
SCALE READING (MM)
0
10
20
30
40
50
60
70
80
90
100

R-11 FLOW METER CALIBRATION SHEET

FLOW METER NUMBER F5 FLOAT #5
DATE 24/5/84

Ambient Temperature: 0 De .C, 32 Deg.F
Ambient Pressure: 29.46 in.Hg.
R-11 Density: 95.781 Lbs/ft3, 1534.3 Kg/m3

All flow meter readings taken at top of float ball!

FLOW METER READING (mm) GLASS FLOAT	FLOW METER READING (mm) BRASS FLOAT	TIME NEEDED TO PASS .7 Lbs R-11 (s)	FLOW RATE CALCULATED (ml/min)
20		409.77	30.30
50		177.17	70.09
75		119.08	104.27
100	1	88.78	139.86
	20	49.88	248.94
	50	29.66	418.64
	75	21.91	566.73
	100	17.22	721.08

R-11 FLOW METER PRESSURE DROP DATA

FLOW METER NUMBER F5
DATE: 24/5/84

Ambient temperature: 10 Deg.C
R-11 Density: 1510 Kg/m3

FLOW METER READING (mm) GLASS	BRASS	PRESSURE DIFFERENCE (mm R-11)	PRESSURE DROP CALCULATED ACROSS FLOW METER (Pa)
20		5	74.1
50		15	222.2
75		25	370.3
100	1	29	429.6
	20	44	651.8
	50	90	1333.2
	75	154	2281.2
	100	195	2889.6

****

#	X	Y	CALC.Y
1	20	30.3	29.6853464
2	50	70.09	70.7360035
3	75	104.27	104.944884
4	100	139.86	139.153765

Y=BO +B1X
WHERE BO=2.31824161
B1=1.36835524

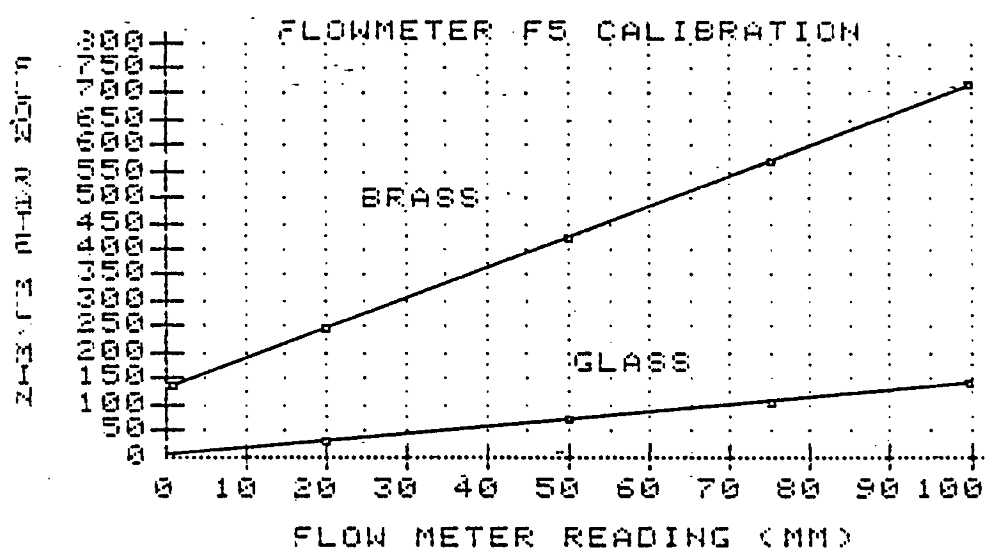
CORRELATION OF FIT.R=.999867263
(R=1 IS A PERFECT FIT)

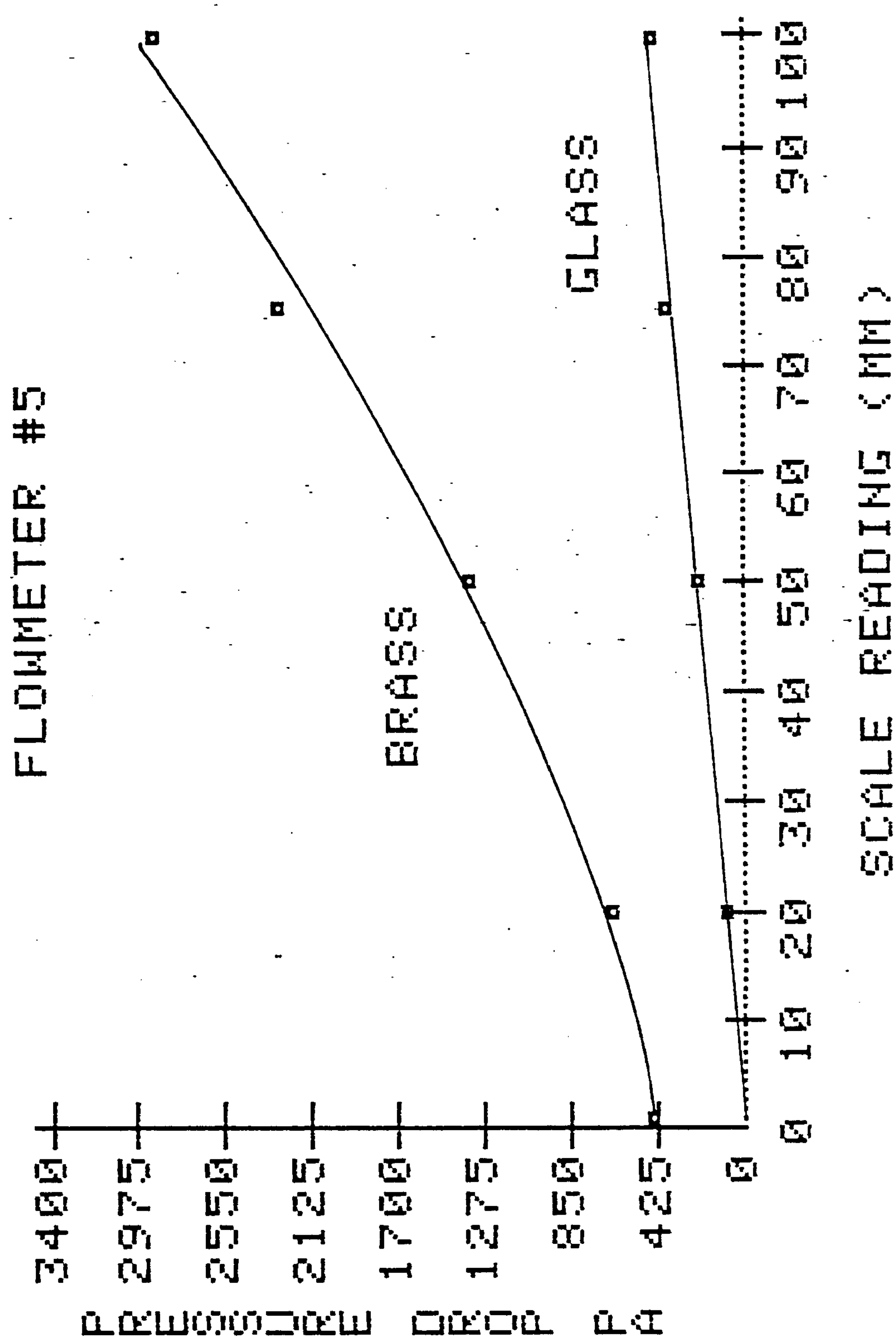
****

#	X	Y	CALC.Y
1	1	139.86	137.052009
2	20	248.94	248.213043
3	50	418.64	423.730465
4	75	566.73	569.994983
5	100	721.08	716.259502

Y=BO +B1X
WHERE BO=131.201428
B1=5.85058073

CORRELATION OF FIT.R=.999844873
(R=1 IS A PERFECT FIT)




FLOWMETER #5
BRASS
GLASS
SCALE READING (MM)
PRESSURE DROP PA
0 425 850 1275 1700 2125 2550 2975 3400
0 10 20 30 40 50 60 70 80 90 100

R-11 FLOW METER CALIBRATION SHEET

FLOW METER NUMBER F6 FLOAT #8
DATE 24/5/84

Ambient Temperature: 8 Deg.C, 46.4 De .F
Ambient Pressure: 29.45 in.Hg.
R-11 Density: 94.664 Lbs/ft3, 1516.4 Kg/m3

All flow meter readings taken at top of float ball!

FLOW METER READING (mm)		TIME NEEDED TO PASS .7 Lbs R-11 (s)	FLOW RATE CALCULATED (ml/min)
GLASS FLOAT	BRASS FLOAT		
20		399.59	31.44
50		181.58	69.19
75	4	126.66	99.19
100	10	88.14	142.53
	20	65.36	192.21
	50	33.45	375.57
	75	24.04	522.58
	100	17.97	699.11

R-11 FLOW METER PRESSURE DROP DATA

FLOW METER NUMBER F6
DATE: 24/5/84

Ambient temperature: 11 Deg.C
R-11 Density: 1508 Kg/m3

FLOW METER READING (mm)		PRESSURE DIFFERENCE (mm R-11)	PRESSURE DROP CALCULATED ACROSS FLOW METER (Pa)
GLASS	BRASS		
20		5	73.9
50		25	369.8
75	4	26	384.6
100	10	28	414.2
	20	29	429.0
	50	75	1109.5
	75	124	1834.4
	100	195	2884.7

****

#	X	Y	CALC.Y
1	20	31.44	29.1813854
2	50	69.19	70.2040142
3	75	99.19	104.389538
4	100	142.53	138.575062

Y=BO +B1X
WHERE BO=1.83296616
B1=1.36742096

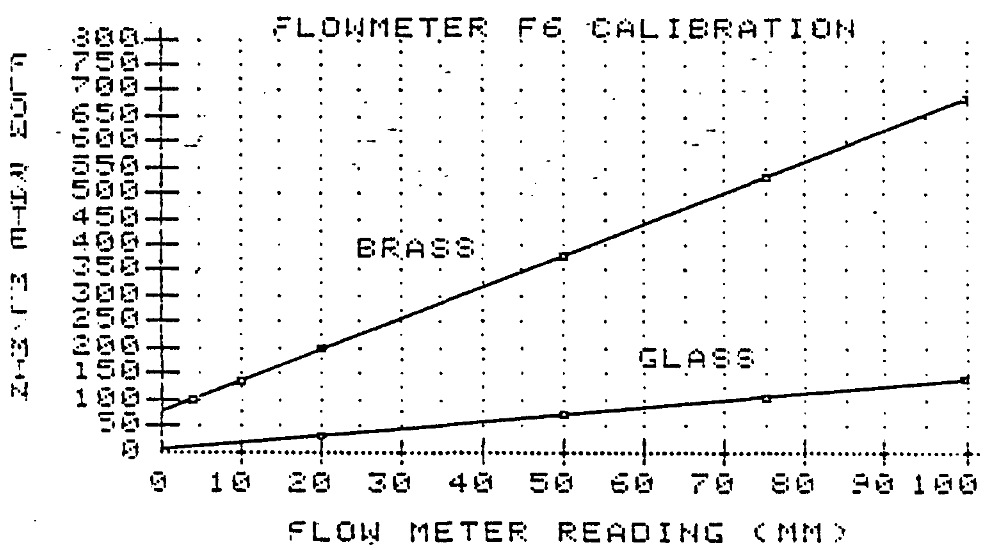
CORRELATION OF FIT.R=.996311536
(R=1 IS A PERFECT FIT)

****

#	X	Y	CALC.Y
1	4	99.19	97.7331364
2	10	142.53	134.621422
3	20	192.21	196.101898
4	50	375.57	380.543325
5	75	522.58	534.244515
6	100	699.11	687.945704

Y=BO +B1X
WHERE BO=73.1409461
B1=6.14804758

CORRELATION OF FIT.R=.999353037
(R=1 IS A PERFECT FIT)

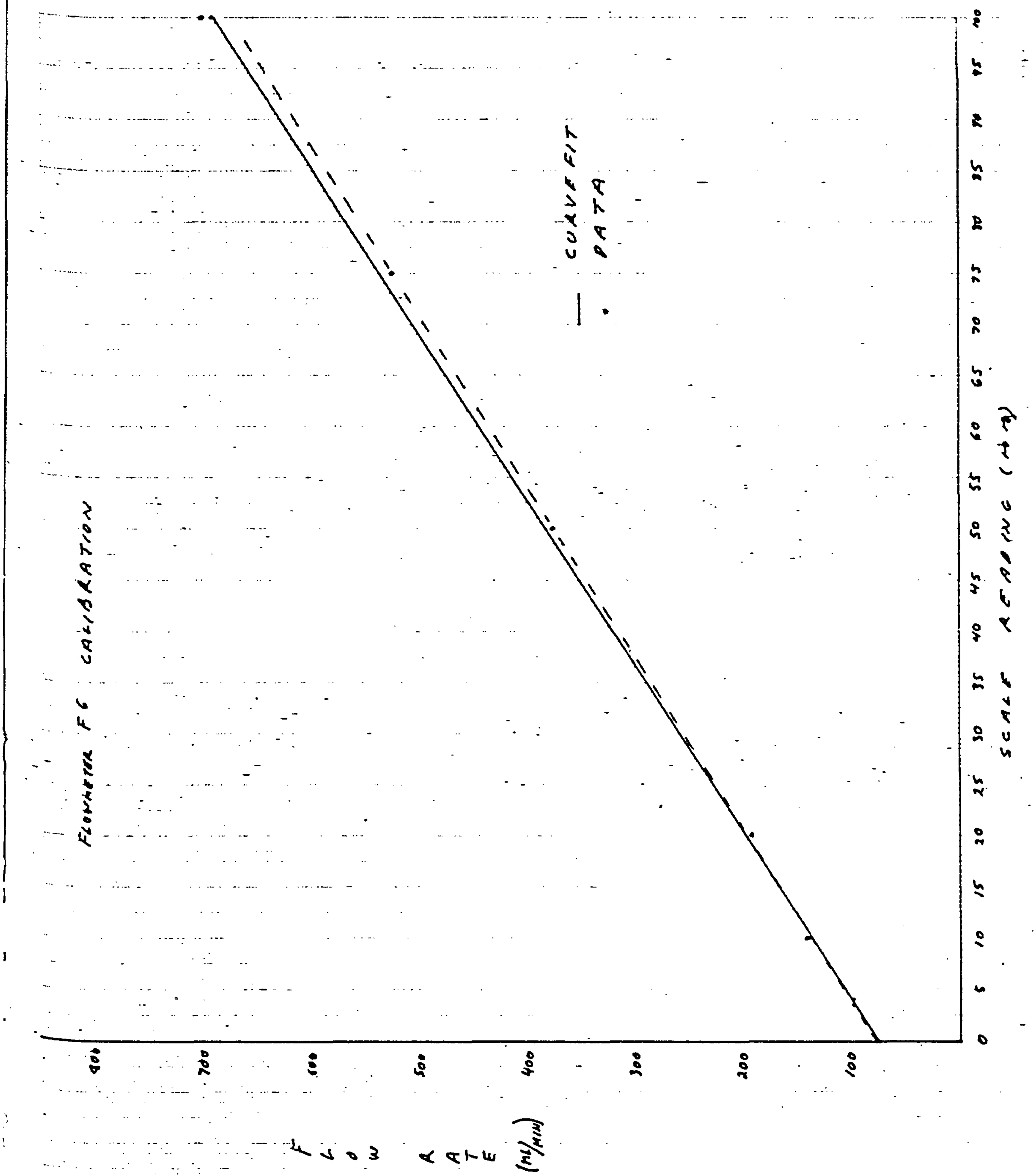


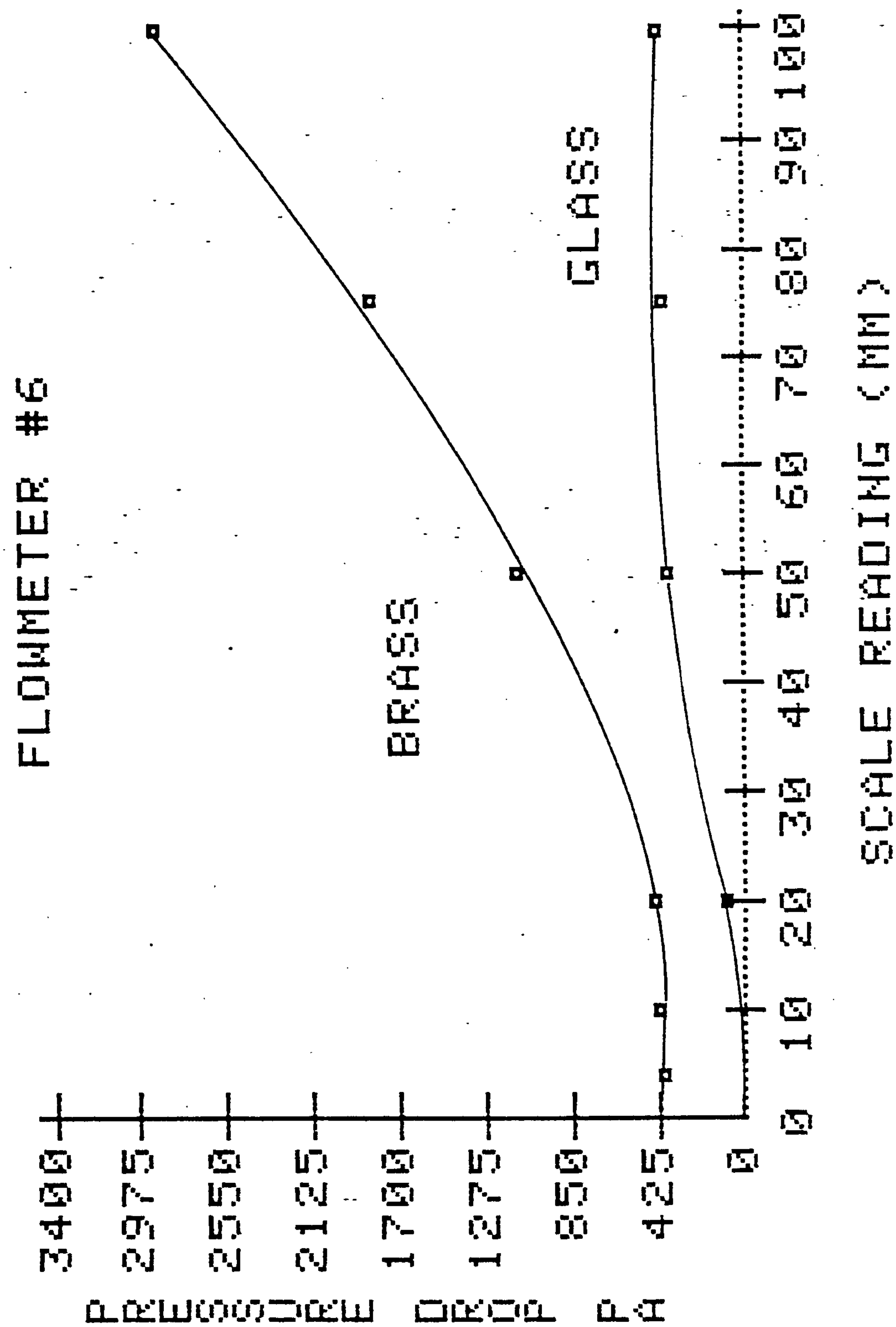
****

#	X	Y	CALC.Y
1	4	99.19	101.404947
2	10	142.53	137.018843
3	20	192.21	196.375337
4	50	375.57	374.444819
5	75	522.58	522.836054

Y=B0 +B1X
WHERE B0=77.662349
B1=5.9356494

CORRELATION OF FIT.R=.999786446
(R=1 IS A PERFECT FIT)


FLOWMETER F6 CALIBRATION
— CURVE FIT
• DATA
SCALE READING (MM)
FLOW RATE (ML/MIN)
0 5 10 15 20 25 30 35 40 45 50 55 60 65 70 75 80 85 90 95 100
100 200 300 400 500 600 700 800


FLOWMETER #6
BRASS
GLASS
SCALE READING (MM)
PRESSURE DROP PA
0
425
850
1275
1700
2125
2550
2975
3400
0
10
20
30
40
50
60
70
80
90
100

R-11 FLOW METER CALIBRATION SHEET

FLOW METER NUMBER F 7 FLOAT #7
DATE 24/5/84

Ambient Temperature: 10 Deg.C, 50 Deg.F
Ambient Pressure: 29.48 in.Hg.
R-11 Density: 94.347 Lbs/ft3, 1511.3 Kg/m3

All flow meter readings taken at top of float ball!

FLOW METER READING (mm) GLASS FLOAT	FLOW METER READING (mm) BRASS FLOAT	TIME NEEDED TO PASS .7 Lbs R-11 (s)	FLOW RATE CALCULATED (ml/min)
20		405.73	31.07
50		173.64	72.60
75		116.83	107.90
100	5	90.58	139.17
	20	57.26	220.15
	50	32.17	391.85
	75	23.44	537.78
	100	18.25	690.72

R-11 FLOW METER PRESSURE DROP DATA

FLOW METER NUMBER F7

DATE: 24/5/84

Ambient temperature: 6 Deg.C

R-11 Density: 1519 Kg/m3

FLOW METER READING (mm) GLASS	BRASS	PRESSURE DIFFERENCE (mm R-11)	PRESSURE DROP CALCULATED ACROSS FLOW METER (Pa)
20		5	74.5
50		17	253.3
75		27	402.3
100	5	28	417.2
	20	35	521.5
	50	75	1117.6
	75	102	1519.9
	100	185	2756.8

****

#	X	Y	CALC.Y
1	20	31.07	31.6720427
2	50	72.6	72.4087389
3	75	107.9	106.355986
4	100	139.17	140.303233

Y=BO +B1X
WHERE BO=4.51424518
B1=1.35788987

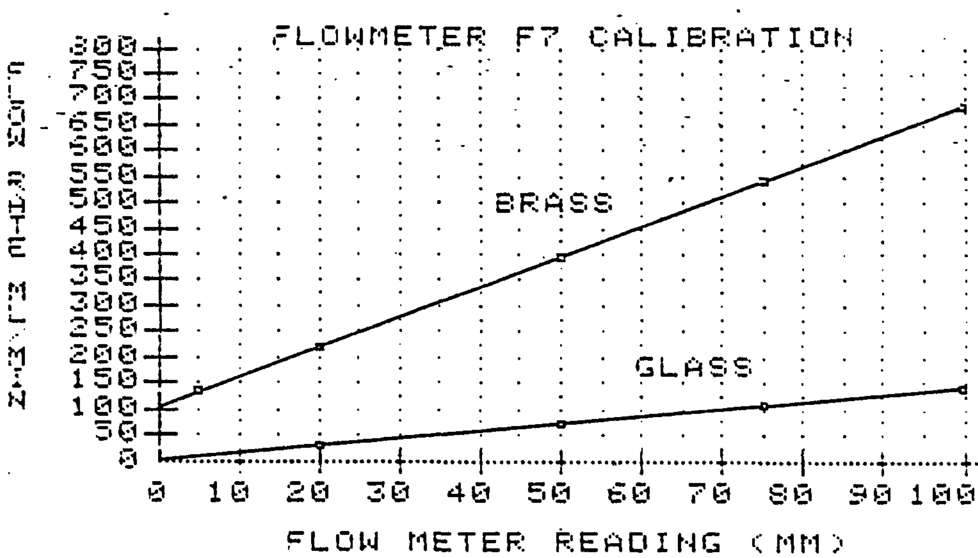
CORRELATION OF FIT.R=.999686705
(R=1 IS A PERFECT FIT)

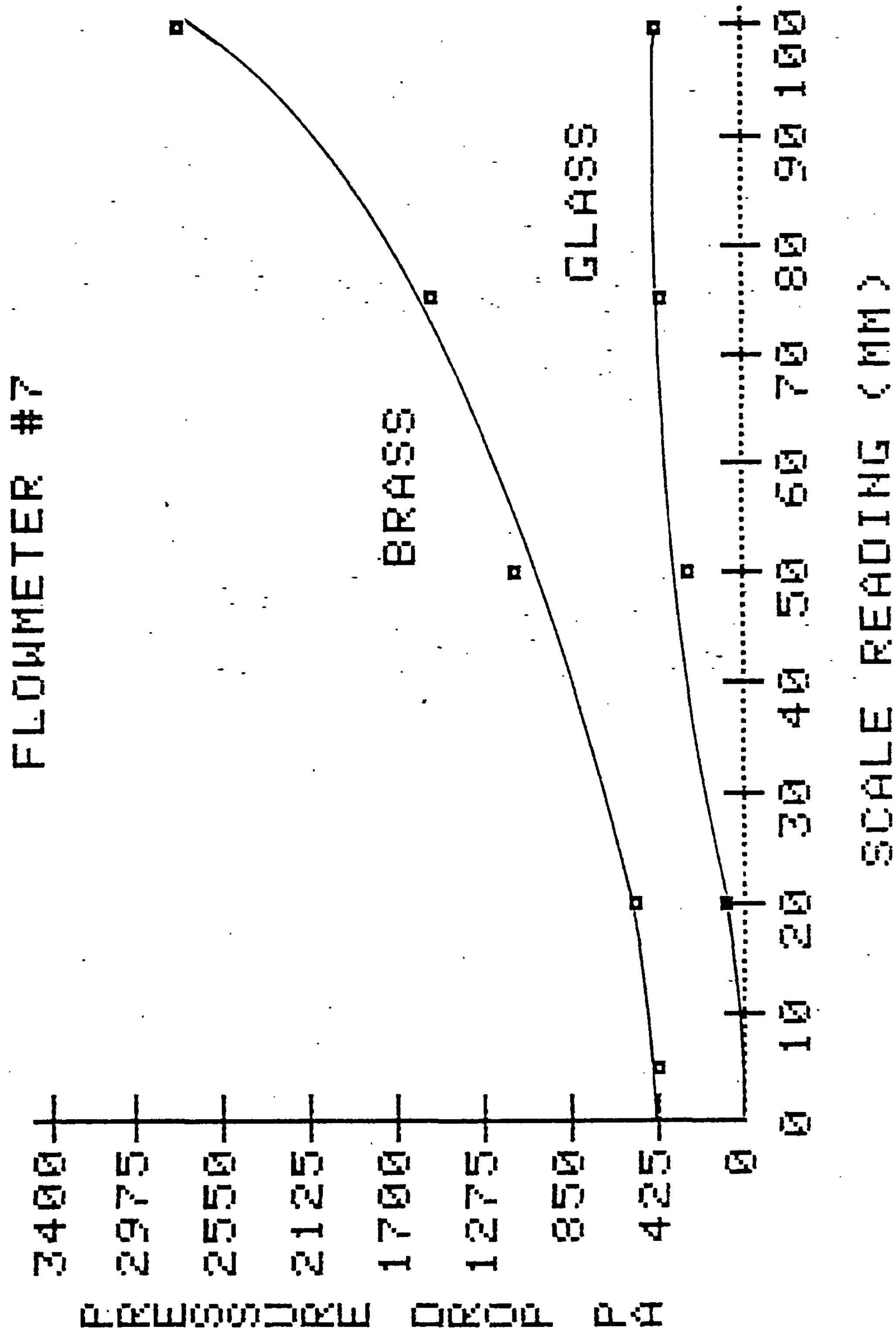
****

#	X	Y	CALC.Y
1	5	139.17	134.760323
2	20	220.15	221.818215
3	50	391.85	395.934
4	75	537.78	541.030487
5	100	690.72	686.126974

Y=BO +B1X
WHERE BO=105.741025
B1=5.80385949

CORRELATION OF FIT.R=.999826899
(R=1 IS A PERFECT FIT)




FLOWMETER #7
BRASS
GLASS
SCALE READING (MM)
0 10 20 30 40 50 60 70 80 90 100
PRESSURE DROP PA
0
425
850
1275
1700
2125
2550
2975
3400

FLOWMETERS*

CALIBRATED & CORRELATED**

a new concept in flow measurement

- **Individually calibrated with air to establish physical constants of meter and yield high precision. *****
- **New correlation based on Stokes' law permits calculation of calibration curve for any fluid for which only the density and viscosity are required.**
- **Each meter serialized and supplied with calibration curve for air and water. Special coordinates convert this curve into a straight line.**
- **Specially designed Teflon® stops allow standard taper joints with precision bore ends to be attached to meter. O-ring seal makes this connection vacuum tight.**
- **Permanent ceramic scale and white background makes reading of black glass ball easy and accurate.**
- **Fluid comes in contact only with glass and Teflon—ideal for corrosive fluids.**

These GILMONT FLOWMETERS are manufactured to ultimate tolerances which give maximum precision for spherical float flowmeters. They are available in five sizes to cover the full range of flows normally encountered with spherical floats. The first three sizes are plain taper tubes while the last two sizes are beaded tubes to give stability where needed.

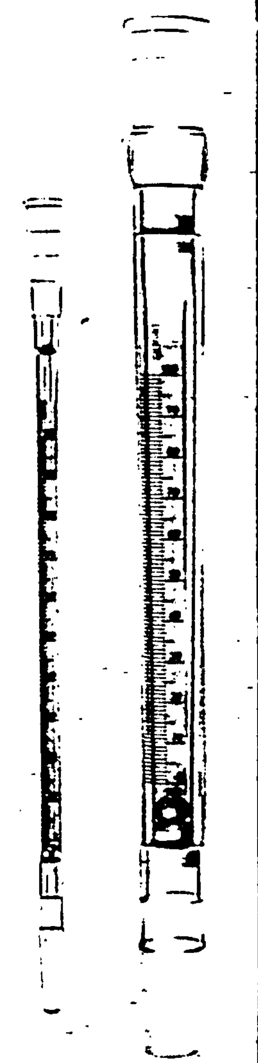
The new theory of correlation based on a dimensionless combination of Stokes' law and flow coefficients makes it relatively simple to precisely calculate the calibration for any fluid. Only the density and viscosity of the fluid at the conditions of flow are required. Because each meter is individually calibrated and plotted with coordinates to give a straight line, highest precision is possible at reasonable price. Each serialized meter comes with its own calibration curve giving flows for air and water directly without any calculation.

Precision-bore ends of each meter combined with special Teflon stops makes it a simple job to disassemble and clean. Standard taper joints with special precision-bore ends can be purchased separately at any time and simply added to the plain tube ends whenever desired.

*U.S. Patent No. 3,183,713 ® T.M. Du Pont

**Reference: Gilmont & Maurer Instr. & Control Sys. 34, 2070 (1941)

***Statistical method of sampling controls accurracy to within ±2% or ±1 scale division (whichever is the greater).



F-1100 F-1500

Shown with respective joint sets attached.

DIRECTIONS

The individual calibration chart supplied with each meter is a plot of the percentage change in diameter ratio, *R*, against the scale reading of the meter, which is a straight line. On the left side of the chart is a direct reading scale of the equivalent flow of air at standard conditions, and a similar scale for water is on the right side.

When cleaning the meter, great care should be exercised not to lose the glass ball. It is possible to replace the glass ball; however, the calibration will not be as accurate, although deviations will only be minor.

Accuracy is ± 2% of reading or ± 1 division, whichever is the greater.

The dimensionless correlation chart *page 7* of flow coefficient, C_R, against Stokes number, S_t, for different values of *R* may be used to calculate the calibration curve for any fluid whose density and viscosity are known at conditions ***of flow, by the following steps:***

1. Select a suitable value of *R* and read the corresponding scale reading from the individual calibration curve supplied.

2. Obtain the values of weight of float in grams, W_f, and the density of float, ρ_f, in gram/ml. from the calibration chart. Let ρ = density of fluid in grams/ml. and μ = viscosity of fluid in centipoises at conditions of flow.

3. Calculate the Stokes number from the following equation:

$$S_t = 1.042 \frac{W_f (\rho_f - \rho)\rho}{\mu^2 \rho_f} R^3$$

4. From the correlation chart read off the corresponding value of C_R for the calculated value of S_t and the selected value of *R*.

5. Obtain the value of diameter of float in inches, D_f, from the calibration chart, and calculate the volumetric rate of flow in ml/min, *q*, from the following equation:

$$q = 59.8\, D_f\, C_R \sqrt{\frac{W_f (\rho_f - \rho)}{\rho_f\, \rho}}\, R \left[\frac{R}{100} + 2\right] K_f$$

$K_f = 1$ in most cases except as follows:

Fluid	Size No.	K_f
Gas (air)	1	$1 + (R/110)^2$
Liq. (water)	1	$1 - (R/100)$
Liq. (water)	2	$1 - (R/250)$
Gas (air)	3	$1 + (R/70)^2$

6. The volumetric rate of fluid flowing and measured at the conditions of flow may be converted into mass rate of flow by multiplying by the density.

7. To obtain a complete calibration curve several values of *R* may be selected; Gases may be plotted against the standard air scale and liquids against the standard water scale on the calibration chart to give very nearly straight lines.

SAMPLE CALCULATION

Flowmeter, catalog No. F 1100, std. water at $R = 25$

1. From individual calibration chart, scale reading is 100 at $R = 25$
2. $W_f = .00530$ grams, $\rho_f = 2.53$ grams/ml. For std water: $\rho = 1.0$ grams/ml and $\mu = 1.0$ cp.
3. $S_t = \dfrac{1.042 \times .00530 \times 1.53 \times 1.00}{1.0 \times 2.53} \times 15.625 = 53.2$
4. From correlation chart, $C_R = .458$
5. $D_f = .0625$ inches and
   $q = 59.8 \times .0625 \times .458 \sqrt{\dfrac{.00530 \times 1.53}{2.53 \times 1.0}} \times 25(2.25) \times (.75)$
   $=$ **4.08** ml/min.
6. Mass flow: $4.08 \times 1.0 = 4.08$ grams/min.

JOINT SETS: When used with joint sets, it is recommended that short lengths of flexible tubing (such as tygon or teflon) be used on each end to hold the joint securely to the flowmeter.

SEE APPENDIX FOR REPRINT

FLOWMETERS, PLAIN ENDS / SET OF JOINTS†

SIZE	AIR	WATER	CAT.NO.	PRICE	FLOAT DIAM	TUBE O.D.	TUBE LENGTH	⊤	CAT.NO.	PRICE
1	1-260	.01-4.1	F-1100	Confirm	0625	5/16"	7½"	10/30	F-1121	Confirm
2	10-1900	0.2-36	F-1200		.125	5/16"	7½"	12/30	F-1221	
3	200-12,000	3-290	F-1300	Latest	.250	7/16"	7½"	14/35	F-1321	Latest
4	1000-36,000	10-850	F-1400		.375	11/16"	9"	19/38	F-1421	
5	3000-77,000	30-1900	F-1500	Pricing	.500	15/16"	9"	24/40	F-1521	Pricing
Kit: 1-5			F-6500							

Each meter is supplied with complete directions and correlation chart for calculating the calibration curve for any fluid whose density and viscosity are known. Also included is a S.S. Float. See p. 7.

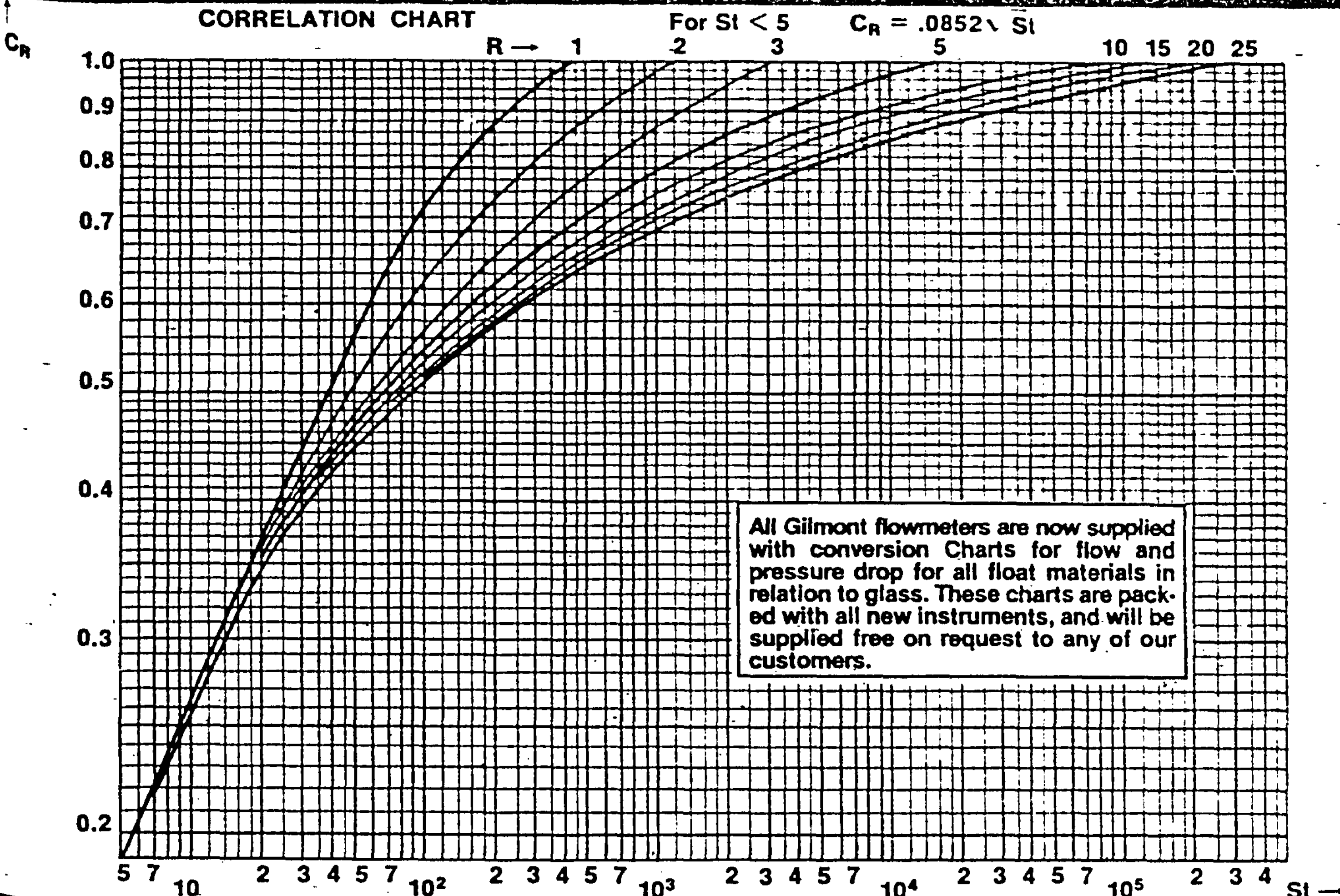
†Consists of one inner and outer joint plus two O-rings

††Flow ranges stated above may be extended by factors of from 2 to 3 by using heavier floats. See Spare Parts List.

NOTE: Extremely dry gases, at low flows, may cause erratic readings due to electrostatic charge build-up. See Microflowmeter, page 11 for polonium ionizing bar.

SPARE PARTS LIST

Multiplying Factor for Floats (see P.7) Gases	Water	SIZE NO. Description & (ρ_f)	1 Cat. No.	Price	2 Cat. No.	Price	3 Cat. No.	Price	4 Cat. No.	Price	5 Cat. No.	Price
		Teflon Stop, Top	F-1104		F-1204		F-1304		F-2404		F-2504	
		Teflon Stop, Bottom	F-1105		F-1205		F-1305		F-1405		F-1505	
		Flowmeter Tube	F-1131	Confirm	F-1231	Confirm	F-1331	Confirm	F-1431	Confirm	F-1531	Confirm
		Joint, Inner	F-1133		F-1233		F-1333		F-1433		F-1533	
		Joint, Outer	F-1134	Latest	F-1234	Latest	F-1334	Latest	F-1434	Latest	F-1534	Latest
		O-Ring for Joint	S-1105	Pricing	F-7032	Pricing	S-1205	Pricing	F-1435-V	Pricing	F-1535-V	Pricing
1	1	Float, Glass (2.53)	F-1132		F-1232		F-1332		F-1432		F-1532	
1.78	2.14	Float, 316 S.S. (8.02)	F-1132-S		F-1232-S		V-2119-S		F-1432-S		F-1532-S	
2.56	3.19	Float, Tantalum (16.6)	F-1132-T		F-1232-T		F-1332-T					
2.43	3.01	Float, Tungsten C. (14.9)							F-1432-TC		F-1532-TC	



ANALYTICAL EQUATION FOR C_R vs St & R

The flow coefficient C_R may be calculated within an error of $\pm 2\%$ to $\pm 5\%$ (depending on the variables) by using the analytical function (common logs):

$$C_R = \frac{\sqrt{b^2 + 4ac} - b}{2a} \quad \text{where}$$

$a = 3.08 \log R - 1.25$

$b = 3.83 - 1.17 \log R$

$c = \log St - .111 \log R$

Example of calculation; interpolating the dimensionless chart of C_R vs St & R for value of R and St with:

$R = 12.8$ and $St = 39.6$, then

$\log R = 1.108$ and $\log St = 1.598$

$a = 3.41 - 1.25 = 2.16$

$b = 3.83 - 1.30 = 2.53$

$c = 1.598 - .123 = 1.475$

$$C_R = \frac{\sqrt{6.41 + 12.78} - 2.53}{2 \times 2.16}$$

$$= \frac{1.85}{4.32} = .428$$

C_R (From Chart) $= .43$

To use the Stokes number and volumetric flow rate equations for gases, ρ and μ at 70°F & 1 atm. must be known.

ρ for any gas may be approximated by using the molecular weight in the equation: $\rho^\circ = .0414 \times$ mol. wgt. $\times 10^{-3}$ g/ml. Temp: & Press. corrections to 70°F & 1 atm can be made from:

$$\rho' = \rho^\circ \times \frac{P}{760} \times \frac{530}{T}$$

with P in Torr and T in °R.

Example: to find ρ for O_2 at 70°F & 1 atm $\rho^\circ = .0414 \times 32 \times 10^{-3} = .00132$ g/ml.

Since μ cannot be determined readily, μ for commonly used gases are listed below in Table 1. ***See Appendix for Nomograph pp. 49-51.***

TABLE I

GAS	μ in cp	GAS	μ in cp
Air	.0181	H_2S	.0126
Ar	.0221	CH_4	.0109
CO_2	.0148	CH_3Cl	.011
CO	.0174	Ne	.031
Cl_2	.0133	NO	.0188
C_2H_4	.0101	N_2	.0175
He	.0194	N_2O	.0144
H_2	.0087	O_2	.020
HCl	.0144	SO_2	.0125
HI	.0166	Xe	.0226

Density & Viscosity of Liquids at 70°F & 1 atm are listed in Table II:

TABLE II

LIQUID	ρ in g/ml	μ in cp
Acetic Acid	1.049	1.221
Acetone	.790	.32
Aniline	1.022	4.40
Benzene	.879	.652
Butyl Acetate	.883	.732
n-butanol	.810	2.948
CCl_4	1.594	.969
Chlorobenzene	1 106	.799
Chloroform	1.483	.58
Diethyl Ether	.714	.233
Ethyl Acetate	.900	455
Ethanol	.789	1.20
Ethylene Br	1.460	1.72
Ethylene Cl	.898	.79
Ethylene Glycol	1.109	19.90
Fluorobenzene	1.023	.598
Heptane	.684	.409
Methyl Acetate	.933	.381
Methanol	.791	.597
Nitrobenzene	1.204	2.03
n-octane	.703	.542
Propanol	.804	2.256
Toluene	.867	.590
o-xylene	.880	.810
m-xylene	.864	.620
p-xylene	.861	.648
Aqueous Solutions (% by weight)		
10% HCl	1.048	1.159
30% HCl	1.149	1.702
10% HNO_3	1.054	1.042
40% HNO_3	1.247	1.558
10% H_2SO_4	1.066	1.228
60% H_2SO_4	1.499	5.905

FLOW MULTIPLYING FACTOR

The *FLOW MULTIPLYING FACTOR* F is defined by step 5 on P. 6: $F = \frac{q'}{q} = \frac{C_R'}{C_R}\sqrt{\frac{\rho_f' - \rho}{\rho_f - \rho}}$ where the prime refers to variables for the heavier float. When $C_R' \to C_R$, $F = \sqrt{\frac{\rho_f'}{\rho_f}}$ for gases and $F = \sqrt{\frac{\rho_f' - 1}{\rho_f - 1}}$ for water. For other liquids, substitute the density. Maximum accuracy is obtained using C_R' and C_R from steps 3 & 4 on P. 6. Since $\frac{W_f}{\rho_f} = V_f$ (constant), $\frac{St'}{St} = \frac{\rho_f' - \rho}{\rho_f - \rho}$.

SIZE NO. 3

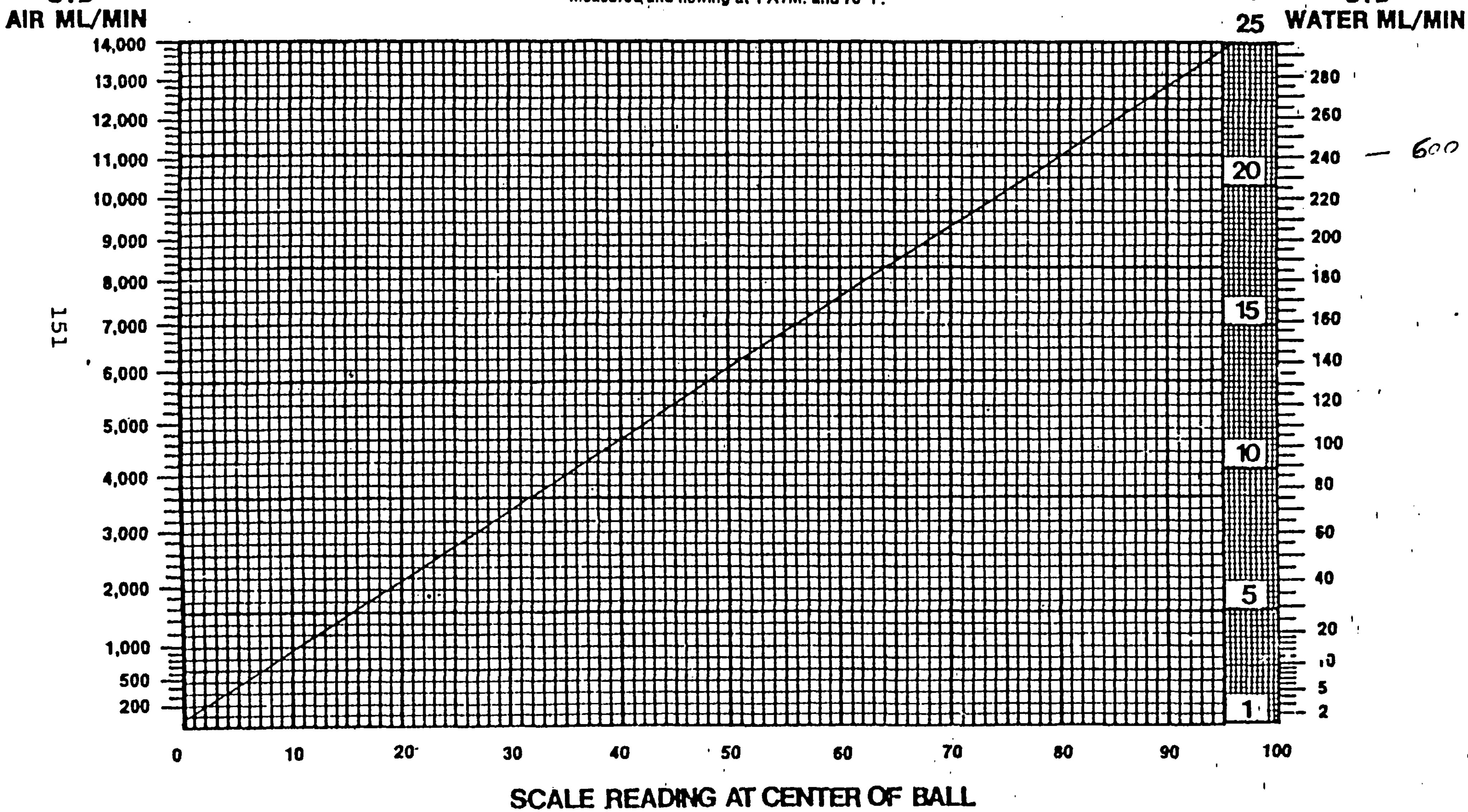
GILMONT INSTRUMENTS, INC. Calibration Chart

Flowmeter Catalog No. F1300

Serial No. C-6351

$D_f = 0.250''$ $W_f = 0.339$ GM $\rho_f = 2.53$ GM/ML

* Measured and flowing at 1 ATM. and 70° F.



APPENDIX C

THERMOCOUPLE INDEX

THERMOCOUPLE INDEX

The index below shows the relationship between the numbering system found on the TSHE apparatus and the one used in the TSHE data acquisition program for the ducting air temperatures measured. The Fluke Data Logger uses the same numbering system as labelled on the apparatus.

EVAPORATOR:	APPARATUS OR FLUKE CHANNEL #			DATA ACQUISITION PROGRAM		
UPSTREAM				TE{I}		
	35	36	37	16	17	18
	32	33	34	13	14	15
	29	30	31	10	11	12
	26	27	28	7	8	9
	23	24	25	4	5	6
	20	21	22	1	2	3
DOWNSTREAM				TF{I}		
	15	16	17	16	17	18
	12	13	14	13	14	15
	9	10	11	10	11	12
	6	7	8	7	8	9
	3	4	5	4	5	6
	0	1	2	1	2	3

CONDENSER:	APPARATUS OR FLUKE CHANNEL #						DATA ACQUISITION PROGRAM					
UPSTREAM							TC{I}					
	77	74	71	68	65	62	18	15	12	9	6	3
	76	73	70	67	64	61	17	14	11	8	5	2
	75	72	69	66	63	60	16	13	10	7	4	1
							TD{I}					
DOWNSTREAM	57	54	51	48	45	42	18	15	12	9	6	3
	56	53	50	47	44	41	17	14	11	8	5	2
	55	52	49	46	43	40	16	13	10	7	4	1

The 16 refrigerant temperatures measured before and after the evaporator were also determined as well as the ambient air temperature and the outside condenser vapour header temperature for row #1. The index below shows the relationship between the numbering system used on the apparatus ,the channel number used in the Fluke Data Logger,and the variable names used in the data aquisition program.

TSHE ROW#	MARKED ON APPARATUS AS:	THERMOCOUPLE CHANNEL # USED	DATA ACQUISITION PROGRAM
Condensate side			TL(I)
1	91	18	1
2	92	19	2
3	93	38	3
4	94	39	4
5	95	58	5
6	96	59	6
7	97	78	7
8	98	79	8
Vapour side			TV(I)
1	80	80	1
2	81	81	2
3	82	82	3
4	83	83	4
5	84	84	5
6	85	85	6
7	86	86	7
8	87	87	8

Thermocouple # 80 - ambient air temperature (TA)

Thermocouple # 89 - outside condenser vapour header temperature (CH(89))

APPENDIX D

VELOMETER CALIBRATION

ALNOR VELOMETER AND PROBE CALIBRATION

As mentioned earlier in Chapter 4 of this thesis, an Alnor Velometer and specially constructed probe was used to determine the air velocity profiles within the ducting at each of the four measurement stations. Due to the static pressure developed within the duct both before {neg.} and after {pos.} the fan, a two port probe was required with the Velometer. Although the Alnor Velometer comes equipped with such a probe whose velocity range and physical size is ideal for this application, it was difficult to seal the probe against the ducting in order to retain the original velocity profile and static pressure once the probe was inserted. To resolve this, a similar probe was fabricated from 1/2 inch copper tubing and brass inserts which could be sealed against the ducting by sliding through a rubber template. Figure D.1 shows the designed probe and the template arrangement used.

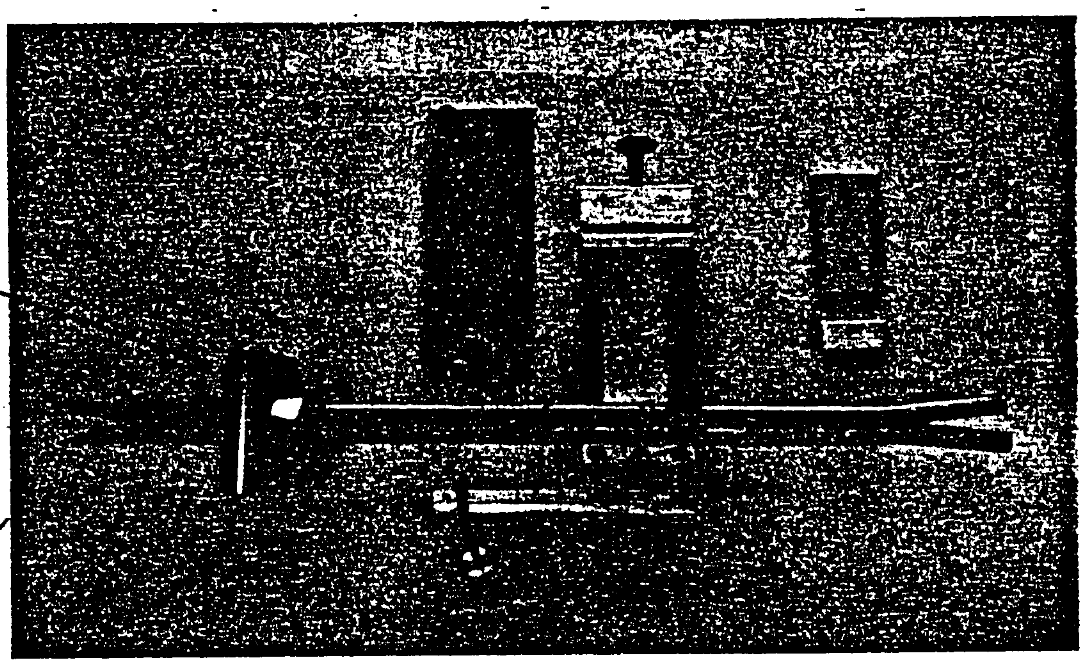


FIGURE D.1
SPECIALLY CONSTRUCTED VELOMETER PROBE
AND TEMPLATE USED TO MEASURE DUCT VELOCITIES

In order to ensure the accuracy of the constructed probe and velometer combination, a calibration was performed against an accurate manometer with pitot tube. As can be seen in Figure D.2, both velometer probe and pitot tube were mounted centrally, and in series inside a variable, low velocity windtunnel (the flow of air travels from left to right). With the use of a variac, the wind tunnel fan speed and thus air velocity, could be set and changed in order to obtain different air flow rates. The velometer was mounted upright and connected to the probe with flexible transparent tubing. The final calibration, which is the one used in the computer data acquisition program, was performed with 5 foot long tube lengths. Since the probe inlet and outlet ports are not perfectly symetrical, care was taken to note the probe's orientation during the calibration. Figure D.3 shows which port of the probe was used to accept air and which

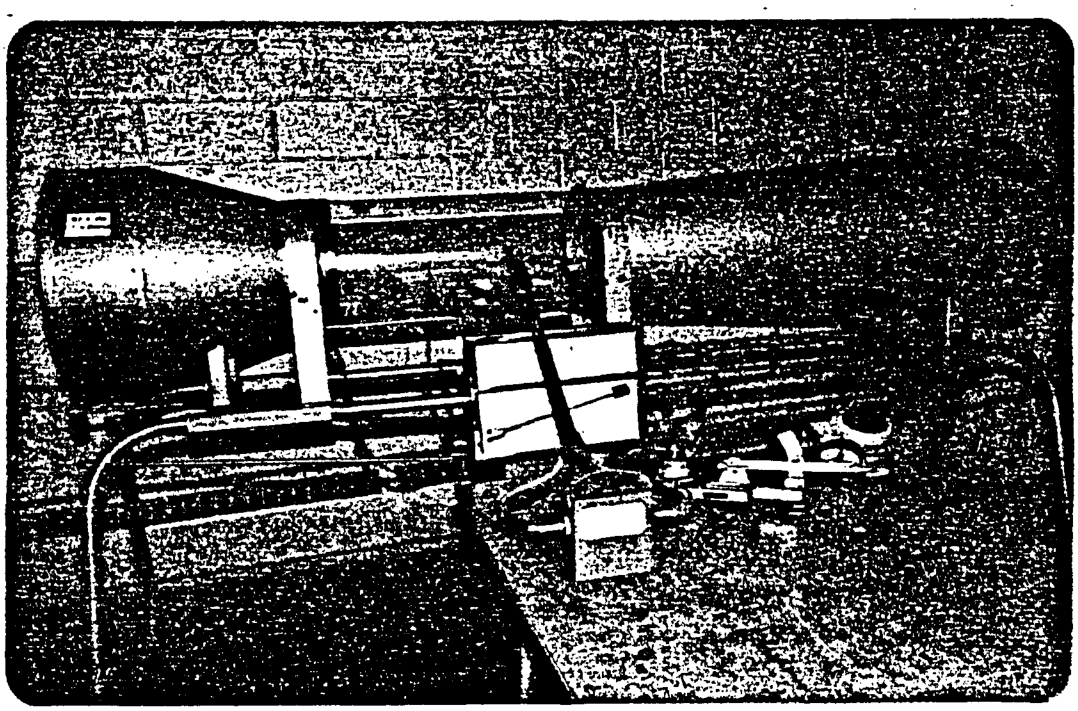


FIGURE D.2
VELOMETER AND PROBE CALIBRATION SETUP

was used to exhaust air.

The manometer was scaled to read 1/25 of a vertical mm of fluid for the Velometer calibration to give good accuracy in the readings , but because of the small pitot tube used, the manometer was slow to respond to changes in the velocity.

CALIBRATION SEQUENCE:

The following list of steps were performed to calibrate the velometer.

1. The apparatus was set up as in Figure D.2
2. Using the variac, the wind tunnel was turned on and the air velocity was increased so that the velometer read almost full scale.
3. After approximately 5 minutes, both a manometer and velometer reading was taken and recorded.
4. The Variac was decreased so as to obtain a slightly slower air flow rate in the windtunnel.
5. Steps 3 and 4 were then repeated until the manometer could no longer be read confidently.

The results of the calibration can be seen in Figure D.4 which shows good agreement between the Manometer and Velometer readings.

A least squares curvefit was performed for velometer readings {FPM} versus manometer reading {m/s} from which equation D.1 was determined.

$$\underset{(M/S)}{\text{Air Velocity}} = .262827199 + 4.81968389E-3 * \underset{(FPM)}{\text{(Velometer reading)}}$$

EQUATION D.1

To accurately place the middle of the probe inlet port at the center of the 1/18 area rectangles in the ducting, the probe position within the mounting fixture was moved for each measurement station. Table D.1 is a summary of the 4 different probe positions within the mounting fixture corresponding to the measurement station used. Note which insert blocks were used and the position along the probe where the fixture was mounted.

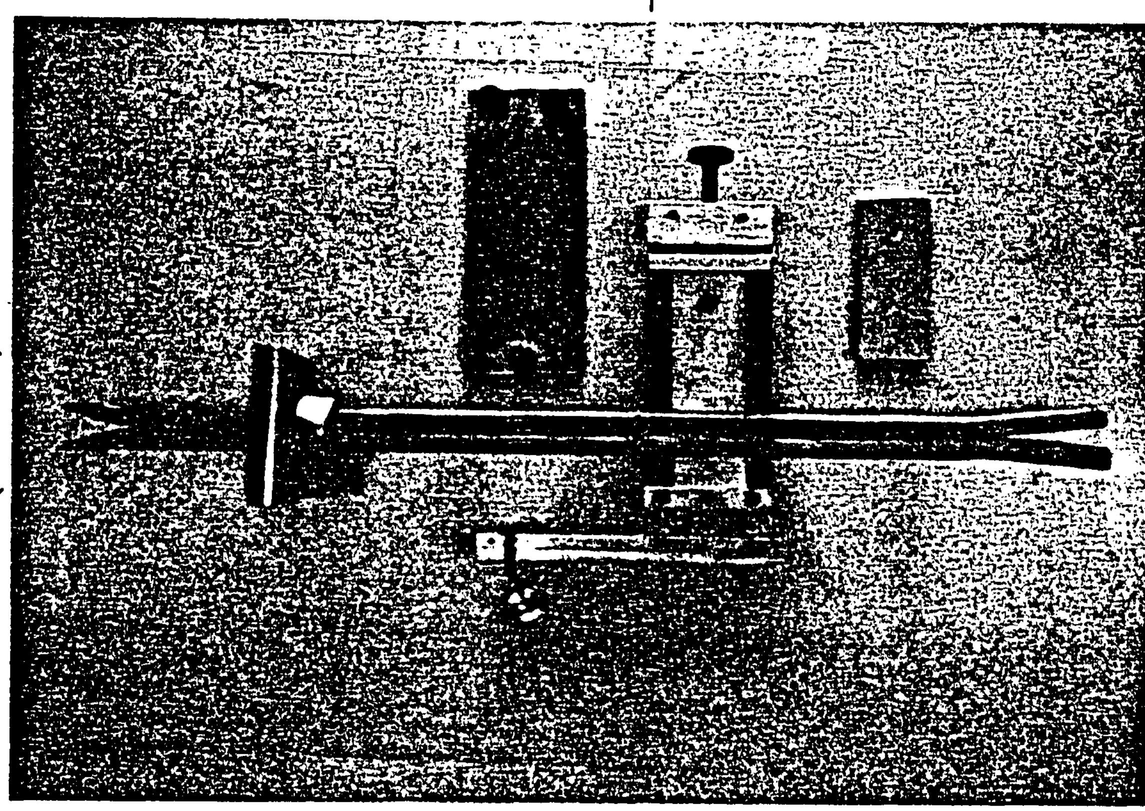


FIGURE D.3
PROBE ORIENTATION IN MOUNTING FIXTURE

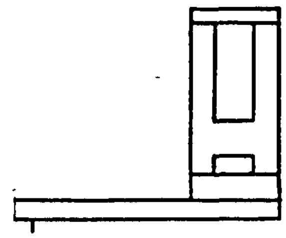
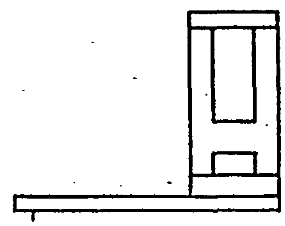
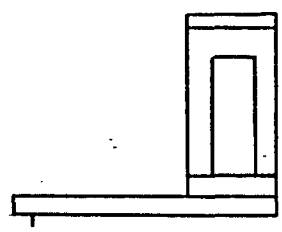
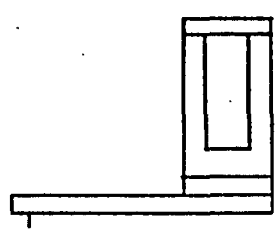
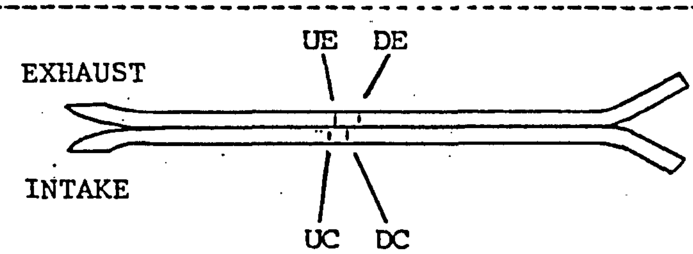
MEASUREMENT STATION	INSERT BLOCKS
UPSTREAM EVAPORATOR	 BLOCK #2 BLOCK #1
DOWNSTREAM EVAPORATOR	 BLOCK #2 BLOCK #1
UPSTREAM CONDENSER	 BLOCK #3
DOWNSTREAM CONDENSER	 BLOCK #3
PROBE POSITIONING MARKINGS	

TABLE D.1
PROBE AND MOUNTING FIXTURE CONFIGURATIONS
FOR DIFFERENT MEASUREMENT STATIONS

VELOMETER CALIBRATION

DATE 17/7/84
Room temp. 24.5 Deg.C
Barometric Pressure 29.284 in.Hg.
Dew Point Temp. 57°F

Humidity Ratio .010118
Man. fluid Specific Gravity .827
Slope of Manometer 1/25

MANOMETER READING (mm)	CALC. FLOW RATE FPM	M/S	VELOMETER READING FPM
42.5	947.9	4.8157	940
32.5	828.98	4.2112	820
23.5	704.91	3.5809	695
15.0	563.18	2.8609	540
5.5	341.02	1.7324	300
1.0	145.41	.73869	100

* VELOMETER HOSE LENGTH: 5 FT.

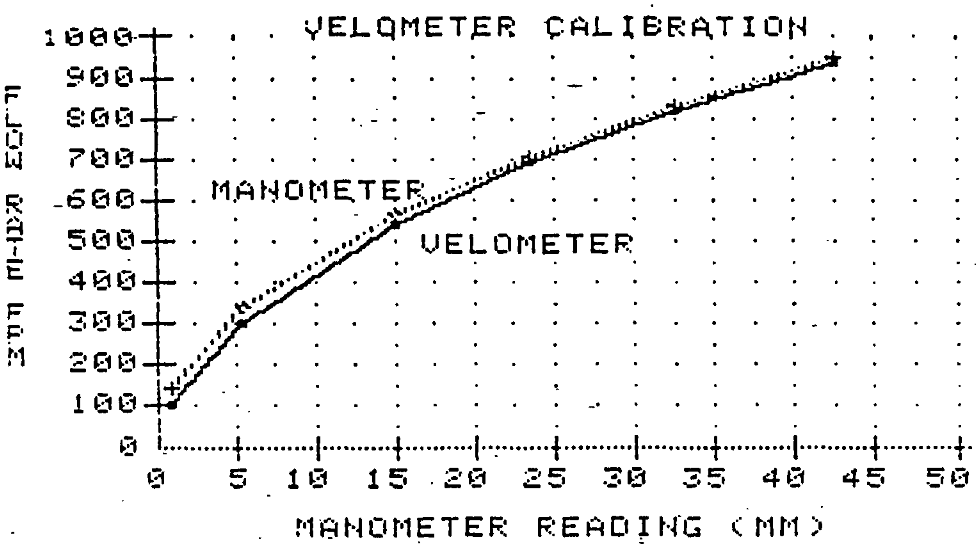


FIGURE D.4
COMPARISON OF VELOMETER WITH MANOMETER READINGS

****

ORIGINAL DATA POINTS

#	X	Y
1	100	.73869
2	300	1.7324
3	540	2.8609
4	695	3.5809
5	820	4.21120001
6	940	4.8157

Y=BO +B1X
WHERE BO=.262827199
B1=4.81968389E-03

CORRELATION OF FIT.R=.999909968
(R=1 IS A PERFECT FIT)

APPENDIX E

SAMPLE RESULTS OF TESTS

RESULTS OF TEST # 7BD

****

1. DATE:11/12/84
2. TIME:14:56:35
2. DATE:11/12/84
3. DAMPER POSITION (DEG.OPEN).....75
4. # OF LOOPS:....................4
5. # OF LOOPS WITH R-11:..........4
6. # OF .375 IN. RECIRC.LINES.....16
7. FAN RPM:.......................1700
8. DEW PT.TEMP.(DEG.C):...........2.22
9. REFER.AIR VELO.(FPM)
   UPSTREAM EVAP.:445 DNSTREAM EVAP.:465
   UPSTREAM COND.:305 DNSTREAM COND.:440
10. EVAP. R-11 CHARGES(IN.)

ROW#	1	2	3	4	5	6	7	8
STATIC	28.88	28.88	29	29	29	29	29.13	29.13
DYNAM.	15.38	20.5	15.88	21.25	14.88	21.5	16.63	19.63
REC.L.	SF	SF	SF	SF	SF	SF	SF	SF

11. DUCT STATIC PRESSURES(MM OF HG)
   UPSTREAM EVAP.=.23 DNSTREAM EVAP.=1.47
   UPSTREAM COND.=-2.7 DNSTREAM COND.=-1.36
12. ROOM BAROMETRIC PRESSURE(in.OF HG)..29.5118257
13. CONDENSER INCLINATION...............45 DEGREES
14. HUMIDITY RATIO......................4.5E-03
15. ROOM TEMPERATURE....................29.6 DEG.C
16. DELTA TEMPERATURE...................39.68 DEG.C
17. AVERAGE STATIC CHARGE...............80.56 %

****

THESE RESULTS ARE BASED ON DUCT AIR VELOCITY AND TEMPERATURE

TOTAL ENERGY ABSORBED BY EVAP............................ 14304.03 WATTS
TOTAL ENERGY EXPELLED BY COND............................ 12482.25 WATTS
TOTAL ENERGY INPUT BY FURNACE............................ 13770.67 WATTS

ENERGY ABSORBED BY EVAP.(1/3 V.SECT. CLOSEST TO R.L.)...... 4640.78 WATTS
ENERGY ABSORBED BY EVAP.(MIDDLE 1/3 V.SECTION)............. 5063.64 WATTS
ENERGY ABSORBED BY EVAP.(1/3 V.SECT.FURTHEST FROM R.L.).... 4778 WATTS

TOTAL ENERGY ABSORBED BY EVAP.USING R-11 MASS FLOW RATES.. 15546.7 WATTS

HEAT LOST THROUGH VAPOUR SIDE OF PIPING.................... 39 WATTS
HEAT LOST THROUGH LIQUID SIDE OF PIPING.................... 29.36 WATTS

SYSTEM THERMAL EFFECTIVENESS............................... 55.59%
SYSTEM TEMP. EFFECTIVENESS................................. 56.68%

AIR FLOW RATES	MASS KG/S	VELOCITY M/S	AVE.TEMP. DEG.C
UPSTRM EVAP.......	.613	2.2	72.57
DNSTRM EVAP.......	.669	2.24	50.08
UPSTRM COND.......	.592	1.89	32.89
DNSTRM COND.......	.636	2.16	52.41

COND. VAPOUR HEADER(ROW #1) OUTSIDE SURFACE TEMP.... 55.5 DEG.C

TEMPERATURE DISTRIBUTIONS IN DUCT (DEG.C)
(ALL TEMP. ARE ARRANGED LOOKING UPRIGHT.DOWNSTREAM)

UPSTREAM OF EVAPORATOR

71.3	71.5	71
70.7	71	70.5
71.7	71.8	72.6
73.3	73.6	73.2
72.8	74.6	74
74.2	74.6	73.8

DOWNSTREAM OF EVAPORATOR

48.4	48.9	49.4
49.3	49.7	49.5
49.9	49.8	49.8
50	50.3	50.2
50.1	50.4	50.6
51.4	51.8	51.9

UPSTREAM CONDENSER

33.3	33	32.8	32.2	31.8	31.5
33.7	33.7	33.7	33.2	32.5	31.8
33.7	33.8	33.8	33.2	32.6	31.7

DOWNSTREAM CONDENSER

52.9	53.2	53.5	53.8	53.8	53.7
52.6	52.8	53.1	53.1	53.2	53.4
51.2	50.9	50.1	50.3	50.1	51.6

R-11 ROW#	MASS FLOW RATES(KG/S)	LIQ.TEMP. DEG.C	VAP.TEMP. DEG.C	ENERGY KJ/S
1	.0155541	54.2	58.7	2671.43
2	.0101413	53.7	58.7	1746.46
3	.0143255	49.4	53.9	2489.87
4	6.8577E-03	48.6	53.9	1196.96
5	.0148027	45.5	48.3	2584.64
6	6.1536E-03	44.9	48.2	1077.52
7	.0125973	42	43.6	2210.29
8	8.9361E-03	41.8	43.6	1569.54

AVERAGE VAPOUR TEMP. DIFFERENCE
BETWEEN LOOPS 1 AND 2: 4.8 DEG.C
BETWEEN LOOPS 2 AND 3: 5.65 DEG.C
BETWEEN LOOPS 3 AND 4: 4.65 DEG.C

****

COMMENTS

ALL FLOATS STABLE (B+-2) WITH LOOP #1 SHOWING THE LARGEST DEVIATIONS.

****

TEST # 788
DATE 11/12/84
BAROMETRIC PRESSURE 29.512 in. Hg
DEW POINT 36.0 Deg.F

R-11 LIQUID CHARGE HEIGHT {IN.}

ROW #	STATIC	DYNAMIC	RECIR.LINE (NF,LF,F,SF)
1	28.88	15.38	SF
2	28.88	20.50	SF
3	29.0	15.88	SF
4	29.0	21.25	SF
5	29.0	14.88	SF
6	29.0	21.5	SF
7	29.13	16.67	SF
8	29.13	14.63	SF

FLOWMETER #	G or B	HEIGHT OF FLOAT mm
1	B	27
2	B	41
3	B	55
4	B	18
5	B	77
6	B	17
7	B	67
8	B	40

" *" denotés exceptionally unstable behaviour

STATIC PRESSURE
(mm of Sg=.827)
UE=
DE=
UC=
DC=

VELOMETER READING {fpm}

UE= 445
DE= 465
UC= 305
DC= 440

COMMENTS: all floats stable (B ± 2) with Comp #1 showing the largest deviations.

APPENDIX F

METHOD OF CALCULATION OF PERFORMANCE AND HEAT LOSS EQUATIONS

METHOD OF TSHE PERFORMANCE CALCULATIONS

This appendix is devoted to a detailed description of the equations and methodology used for the calculation of the performance parameters. The equations shown here along with a nomenclature legend can also be found in Appendix G `Computer Data Acquisition Program `.

```
INPUT BY HAND:  DEW POINT TEMP.       :TW

                BAROMETRIC PRESSURE   :PB

                VELOMETER READINGS

                                  UE : VR(1)
                                  DE : VR(2)
                                  UC : VR(3)
                                  DC : VR(4)

                FAN SPEED             : FS
```

CALCULATIONS:

```
PROGRAM
LINE#          EQUATION                            DESCRIPTION

      ┌FOR I=1 TO 4

2170  │VU(I) = VR(I) * CF(FF,I)                    AVERAGE VELOCITY
      │                                            CORRECTION BETWEEN
      │                                            MEASUREMENT STATIONS
      │                                            FF=1 (1150 RPM)
      │                                              =2 (1700 RPM)
      │                                              =3 (2300 RPM)

2310  │VT(I) = .262827199 + 4.819E-3*VU(I)         VELOMETER CALIBRATION

      └NEXT I
      ┌FOR I = 1 TO 18

2320  │VE(I) =  VT(1)  *  NV(FF,1,I)               VELOCITY AT
      │VF(I) =  VT(2)  *  NV(FF,2,I)               MEASUREMENT STATION
      │VC(I) =  VT(3)  *  NV(FF,3,I)               GIVEN REFERENCE
      │VD(I) =  VT(4)  *  NV(FF,4,I)               AND NORMALIZED
      │                                            VELOCITY PROFILE

2440  │VC(I) =  VC(I) + DV(I,DP)                   CORRECTION FOR
      │                                            DAMPER POSITION
      └NEXT I                                      DP=1,2,3,4, OR 5
```

```
4060     INPUT: FLOWMETER READINGS    FLOAT TYPE(B/G): WW$(I)
                                      HEIGHT(MM)     : MG(I)

4130-4280   FLOWMETER CALIBRATION EQUATIONS USED TO OBTAIN  VOLUME
            FLOW RATE

4340     MR(I) = {(D6-D4)/(D6-D5(I))*            MASS FLOW RATE OF
                  D5(I)/D4}^.5 *MR(I)*D4/6E7     R-11 IN SYSTEM FROM
                                                 LOOP # I
                                                 (R-11 DENSITY
                                                 CORRECTION)

5050     K = C1/TW+C2+C3*TW+C4*TW^2+C5*TW^3+     USED IN LINE 5060
             C6*LOG(TW)                          FOR SATURATION
                                                 PRESSURE OF WATER
                                                 81 ASHRAE FUND.
                                                 HANDBOOK,PG 5.2,
                                                 EQN. 4
                                                 C1-C6 ARE CONSTANTS

5060     PS = EXP(K)                             SATURATION PRESSURE
                                                 OF WATER VAPOUR

5080     W = .62198 * PS/(P-PS)                  HUMIDITY RATIO
                                                 P-BAROMETRIC PRES.
                                                 81   ASHRAE   FUND.
                                                 HANDBOOK Pg.5.3,EQN.21

        ┌FOR J = 1 TO 18
        │
5130    │HE(J) = TE(J)+W*(2501+1.805*TE(J))      ENTHALPY OF DRY AIR
        │                                        81 ASHRAE FUND.
        │                                        HANDBOOK,Pg.5.3 EQN 27
        │
        │DE(J) = (PB+PE-PS)/(287.055*(TE(J)+273))
        │                                        DENSITY OF DRY AIR
        │                                        81    ASHRAE   FUND.
        │                                        HANDBOOK,Pg.5.3 EQN 29
        │
5180    │ME(J)  = DE(J) * VE(J) *AR              MASS FLOW RATE OF
        │                                        AIR FOR EACH 1/18
        │                                        AREA
        │
        │J1     = HE(J) * ME(J) + J1             TOTAL HEAT ENERGY
        │                                        FOR MEASUREMENT
        │                                        STATION #1
        │
        │M1     = ME(J) + M1                     TOTAL MASS FLOW
        │                                        RATE FOR MEASUREMENT
        │                                        STATION #1
        └NEXT J
```

```
5225   H1 = J1 / M1                            AVERAGE HEAT FLUX
                                               FOR MEASUREMENT
                                               STATION #1

5230   E1 = (M1 + M2)/2   * (H1-H2)            ENERGY ABSORBED BY
                                               EVAPORATOR

       E2 = (M3 + M4)/2   * (H4-H3)            ENERGY ABSORBED BY
                                               CONDENSER

       E3 = (M1 + M4)/2   * (H1-H4)            ENERGY INPUT BY
                                               FURNACE

       E4 = (M1 + M3)/2   * (H1-H3)            MAXIMUM AVAILABLE
                                               ENERGY IN SYSTEM
```

HEAT LOSS CALCULATION:

5350 - 5610 SEE THE REMAINDER OF THIS APPENDIX FOR A DESCRIPTION OF THE EQUATIONS USED IN THE HEAT LOSS CALCULATIONS

EFFECTIVENESS:

```
5650      EF    =    (E1+E2)/2/E4*100       THERMAL EFFECT.

          ET    =    (T1+T2)/(T1-T3)*100    TEMP. EFFECT.
```

ENERGY EQUATIONS USING FLOWMETER DATA.

```
5310  HV(I) = BO+B1*(TV(I)+273)+B2*(TV(I)+273)^2*1000
                                               ENTHALPY  OF  VAPOUR
                                               FOR LOOP `I`

5320  HL(I) = DO+D1*(TL(I)+273)+D2*(TL(I)+273)^2*1000
                                               ENTHALPY OF LIQUID
                                               FOR LOOP 'I'
      (SEE THE END OF THIS APPENDIX FOR ENTHALPY CURVEFITS)

5330  QR(I) = MR(I) * (HV(I) - HL(I))          HEAT GAINED BY
                                               EVAP. IN LOOP 'I'

5335  QT   =  QR(I) + QT                       TOTAL HEAT GAINED BY
                                               EVAPORATOR
```

DUCT AND PIPING HEAT LOSS EQUATIONS

To reduce the error in the calculation of the energy absorbed and expelled by the evaporator and condenser respectively, the heat losses through the ducting walls between the measurement stations and the respective heat exchanger coils were accounted for. Since the heat exchanger coil face tempertures were not known, the method described in the ASHRAE 81 Fundamentals Handbook, Pg 33.9 , for estimating the heat loss through a section of ducting was used . Presented below is a synopsis of those equations and the nomenclature used.

Also included in this Appendix is a summary of the methodology and assumptions made in order to calculate the heat lost from the interconnecting piping between the two heat exchangers. Unlike the ducting heat loss calculations, the piping heat losses are not included in the determination of the energy absorbed or expelled by the heat exchangers but instead, are printed alongside of the evaporator and condenser energies so that accountability can be made for the differences between the two.

DUCT HEAT LOSS CALCULATIONS

From page 33.9 of the ASHRAE Fundementals;

$$QL = \frac{UPL}{CF}\left[\frac{Te + Tl}{2}\right] - TA$$

$$Te = \frac{Tl\ (y+1) - 2TA}{(y-1)}$$

$$Tl = \frac{Te\ (y-1) + 2TA}{(y+1)}$$

$$y = CF1 * A * V * \rho / (U*P*L)$$

WHERE:

QL : HEAT LOSS/GAIN THROUGH DUCT WALLS (W)

U : OVERALL COEF. OF HEAT TRANSFER W/M2C)

P : PERIMETER OF DUCT (MM)

Te : TEMP. OF AIR ENTERING DUCT (DEG.C)

Tl : TEMP. OF AIR LEAVING DUCT (DEG.C)

TA : TEMP. OF AMBIENT AIR (DEG.C)

CF : CONVERSION FACTOR = 1000

CF1 : CONVERSION FACTOR = 2.01

A : CROSS SECTIONAL DUCT AREA (MM)

L : LENGTH OF DUCT (MM)

V : VELOCITY OF AIR (M/S)

ρ : DENSITY OF AIR (KG/M3)

FROM FIGURE 8, PG 33.10 FOR AN UNINSULATED DUCT,

VELOCITY (M/S)	U (W/M2C)
0	3.3
5	3.6
10	3.9
15	4.15
20	4.45

SOLVING FOR EQUATION OF BEST FIT (LEAST SQUARES METHOD)

U= 3.31 + 0.057 * (VELOCITY)

ADDITION OF INSULATION TO OUTSIDE OF DUCT WILL ADD APPROX. R-6:

6 FT2 F H/ Btu = 1.06 M2 K/W

THEREFORE: U1 = 1/R + 1/R1

= 1/1.06 + U

DUCT DIMENSIONS	:	HEIGHT	= 3 FEET = 914.4 MM
		DEPTH	= 1 FOOT = 304.8 MM
LENGTH OF DUCT BETWEEN HEAT EXCHANGER AND MEASUREMENT STATION	:	L1	= 12 CM
		L2	= 58 CM
		L3	= 116.84 CM
		L4	= 116.84 CM

LENGTH OF DUCT BETWEEN MEASUREMENT STATION #4 AND FURNACE : L7 = 185 CM
LENGTH OF DUCT BETWEEN MEASUREMENT STATION #1 AND FURNACE : L8 = 191 CM

ENERGY BALANCE: SEE FIGURE E.1 FOR SIGN AND LABEL CONVENTION

1. BETWEEN MEASUREMENT STATION 1 AND 2

EN1 - (EN2 + Q2 + QE + Q1) = 0

QE = EN1 - EN2 - Q2 - Q1

2. BETWEEN MEASUREMENT STATIONS 3 AND 4

EN3 + QC - (Q3 + Q4 + EN4) = 0

QC = EN4 - EN3 + Q3 + Q4

3. BETWEEN MEASUREMENT STATIONS 4 AND 1

QF + EN4 - (EN1 + Q7 + Q8) = 0

QF = EN1 - EN4 + Q7 + Q8

4. MAXIMUM ENERGY ACROSS SYSTEM

QM = ENERGY AT UPSTREAM FACE OF EVAP. - ENERGY AT UPSTREAM FACE OF COND.

= (EN1 - Q1) - (EN3 - Q3)

= EN1 - EN3 - Q1 - Q3

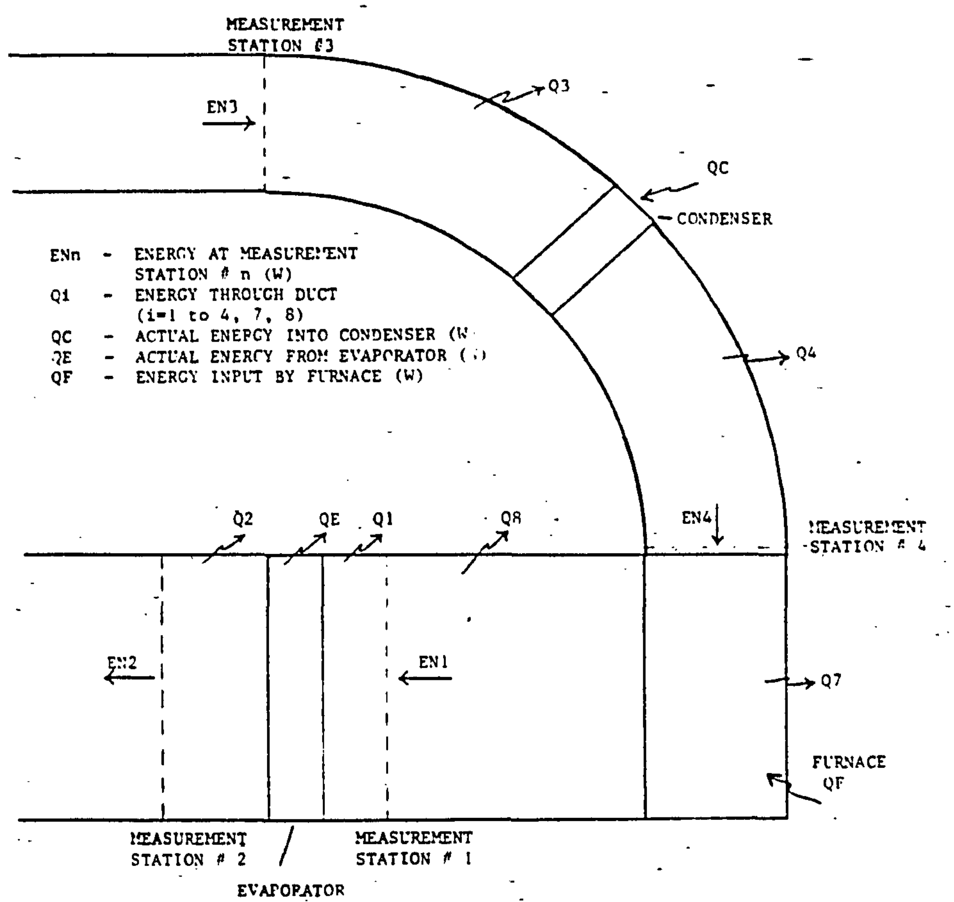


FIGURE F.1

DUCT HEAT LOSS CALCULATIONS SIGN CONVENTION

HEAT LOSS THROUGH PIPING:

THE HEAT LOSS THROUGH THE FLEXIBLE CONDENSATE RETURN TUBING AND INSULATION CAN BE MODELLED BY THE EQUATION:

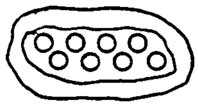
$$Q = \frac{To - Ta}{\{rs \ln(rl/ro)/K2\} + \{rs \ln(rs/rl)/K1\} + Rs} \qquad \text{ASHRAE Pg.23.8}$$

WHERE:
- Q - Rate of heat transfer/sq ft of outer surface of insulation {Btu/hft2}
- K - Thermal conductivity of insulation at mean temp. {Btu in./hr ft2 F}
- L - Thickness of insulation {in}
- Ta - Temperature of ambient air {F}
- To - Temperature of inner surface of insulation {F}
- ro - Inner radius of insulation {in}
- rl - Outer radius of intermediate layers of insulation {in}
- rs - Outer radius of insulation surface {in}
- Rs - Surface resistance = 1/h = FT2 F H/Btu

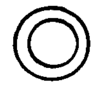
To = TL(I)
rs = 1.1/2 in.
rl = 3/8 /2 in.
ro = .25 /2 in.
K1 = .255 Btu in/Hr FT2 F
K2 = .35 W/M K = .05048 Btu in/Hr FT2 F

HEAT LOSS THROUGH VAPOUR PIPING:

ACTUAL



ASSUMED CONFIGURATION



$Q = U A \Delta T$ ASSUME $R = 6$ Hr FT2 F/Btu

$A = P * t = 1.5 * 13.75$ FT2

$.2930$ W = 1.0 Btu/Hr

$Q = 1/6\ (1.5)\ (13.75)\ (\Delta T)\ (.2930)$

CURVEFIT OF R-11 ENTHALPY VERSUS TEMPERATURE

CURVEFIT OF SATURATED LIQUID DATA

The Temperature versus Enthalpy data for saturated liquid refrigerant R-11 used in the table below was curvefit for use in the computer data acquisition program . The data was obtained from the 1981 ASHRAE Fundamentals Handbook, Table 2, Pg 17.77 . It was determined that a second order polynomial using the least squares error technique for curvefitting yielded the best fit .

POINT #	TEMP.(T) (DEG.K)	ENTHALPY (KJ/Kg)	CALCULATED ENTHALPY(KJ/Kg)
1	270	29.532	29.4999
2	280	38.250	38.2635
3	290	47.102	47.1159
4	300	56.056	56.0574
5	310	65.096	65.0877
6	320	74.217	74.2071
7	330	83.422	83.4154
8	340	92.719	92.7126
9	346	98.347	98.3336

WHERE: HL = B0 + B1(T) + B2(T^2)

B0=-173.489
B1=0.6317230
B2=4.44777E-4

CORRELATION OF FIT: R=.99976

CURVEFIT OF SATURATED VAPOUR DATA

Again, values from Table 2,pg 17.77 of the 81 ASHRAE Fundamentals were used to curvefit Temperature versus Enthalpy data of saturated R-11 vapour by the least squares error technique. The best fit curvefit is shown below.

POINT #	TEMP.(T) (DEG.K)	ENTHALPY (KJ/Kg)	CALCULATED ENTHALPY(KJ/Kg)
1	270	221.23	221.136
2	280	226.40	226.406
3	290	231.58	231.614
4	300	236.73	236.757
5	310	241.83	241.838
6	320	246.88	246.854
7	330	251.84	251.808
8	340	256.69	256.698
9	346	259.53	259.601

WHERE: HV = B0 + B1(T) + B2(T^2)

B0=54.83223
B1=0.701634
B2=-3.17389E-4
CORRELATION OF FIT: R=.99863

CURVEFIT OF R-11 DENSITY VS TEMPERATURE (SATURATED LIQUID)

The following data for the density of saturated liquid R-11 was obtained from the 1981 ASHRAE Fundamentals Handbook, Table 2, Pg. 17.77 . It was found that a second order polynomial using the least squares technique of curvefitting would best represent this data in the temperature range considered. Comparison between table and calculated density values are shown below.

POINT #	TEMP{T} (deg.K)	DENSITY{D} (Kg/m^3)	CALCULATED DENSITY(Kg/m^3)
1	250	1584.8	1584.588
2	275	1529.5	1529.993
3	296.8	1479.4	1479.538
4	300	1471.8	1471.849
5	310	1447.7	1447.599
6	316	1433.0	1432.776
7	320	1423.0	1422.780
8	330	1397.5	1397.393
9	336	1381.8	1381.889
10	340	1371.1	1371.439

where: D = B0 + B1(T) + B2(T^2)

and: B0=1935.35605
B1=-.69333005
B2=-2.8389683E-3

CORRELATION OF FIT: R=.9956

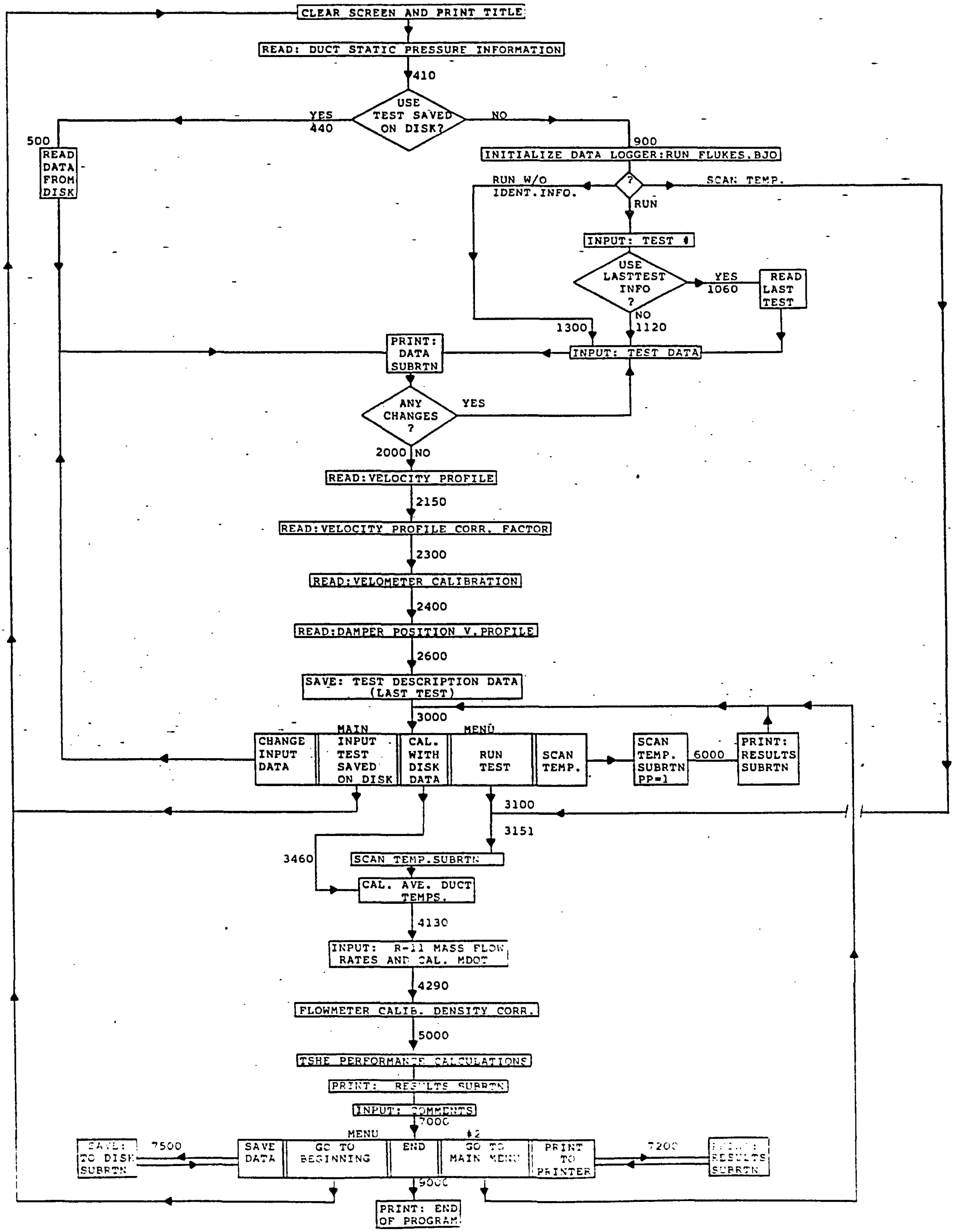
APPENDIX G

COMPUTER DATA ACQUISITION PROGRAM

COMPUTER DATA ACQUISITION PROGRAM

The computer data acquisition program was written to enable fast, error free collection of the information necessary for the calculation of the parameters which indicate the TSHE`s performance. The program was written in Apple Basic and was designed to ask clear, consise questions about the parameters which are require for calculation. It is hoped that almost anyone who wishes to perform an experimental test on the TSHE can do so with little prior knowledge of the calculation methodology involved. The following pages contain the flow diagram, listing of the computer data acquisition program titled `TSHR`, and a nomenclature index for the TSHR program. Also included in this Appendix is a listing of the `Purge` program which should be used whenever the purging of non-condensible gases from the system is required.

The sub-program titled `Flukes.OBJ` is a machine language program which enabled the Fluke`s data transmission to be decoded and stored at specific memory addresses within the Apple Computer. This program is automatically retrieved from disk storage to initialize the Fluke Data Logger whenever the data acquisition program `TSHR` is ran. Special thanks is expressed to Dr.N.Wilson for writing the `Flukes.OBJ` software.



CLEAR SCREEN AND PRINT TITLE
READ: DUCT STATIC PRESSURE INFORMATION
410
USE TEST SAVED ON DISK?
YES
440
NO
500
READ DATA FROM DISK
900
INITIALIZE DATA LOGGER:RUN FLUKES.BJO
?
RUN W/O IDENT.INFO.
SCAN TEMP.
RUN
INPUT: TEST #
USE LASTTEST INFO ?
YES
1060
READ LAST TEST
NO
1120
1300
INPUT: TEST DATA
PRINT: DATA SUBRTN
ANY CHANGES ?
YES
2000
NO
READ:VELOCITY PROFILE
2150
READ:VELOCITY PROFILE CORR. FACTOR
2300
READ:VELOMETER CALIBRATION
2400
READ:DAMPER POSITION V.PROFILE
2600
SAVE: TEST DESCRIPTION DATA (LAST TEST)
3000
MAIN MENU
CHANGE INPUT DATA
INPUT TEST SAVED ON DISK
CAL. WITH DISK DATA
RUN TEST
SCAN TEMP.
SCAN TEMP. SUBRTN PP=1
6000
PRINT: RESULTS SUBRTN
3100
3151
3460
SCAN TEMP.SUBRTN
CAL. AVE. DUCT TEMPS.
4130
INPUT: R-11 MASS FLOW RATES AND CAL. MDOT
4290
FLOWMETER CALIB. DENSITY CORR.
5000
TSHE PERFORMANCE CALCULATIONS
PRINT: RESULTS SUBRTN
INPUT: COMMENTS
7000
MENU #2
SAVE: TO DISK SUBRTN
7500
SAVE DATA
GO TO BEGINNING
END
GO TO MAIN MENU
PRINT TO PRINTER
7200
PRINT: RESULTS SUBRTN
9000
PRINT: END OF PROGRAM

TSHE COMPUTER PROGRAM NOMENCLATURE LEGEND

VARIABLE	DESCRIPTION
A	Variable used in temperature scan routine.
AA	Ambient temperature {deg.F}
AB	Average R-11 Vapour line temp. {deg.F}
AP	Average R-11 Liquid line temp. {deg.F}
AR	1/18 area of duct {m^2}
CF(I,II)	Velocity profile corr. factor,I-fan speed,II-UE,DE,UC,DC
CH{I}	Channel # I from data logger
C1-7	Saturated pressure coefficient used in duct heat loss calc.
DA	Day
DE{I}	Density of dry air upstrm of evap. {Kg/m^3}
DF{I}	Density of dry air dnstrm of evap. {Kg/m^2}
DC{I}	Density of dry air upstrm of cond. {Kg/m^2}
DD{I}	Density of dry air dnstrm of cond. {Kg/m^2}
DP	Damper position #, 5-closed,1-full open
DS{I,J,K}	Duct static pressure {mm Hg}
DV{I,J}	Damper position normalized velocity profile
DZ	Damper percent open
D4	Density of liquid R-11 {Kg/m^3}
D5{I}	Density of R-11 at time of flowmeter{I} calibration.{Kg/m^3}
E1	Total flux across evap. {KJ/s}
E2	Total flux across cond. {KJ/s}
E3	Total flux across furnace {KJ/s}
ED{I}	Evap. dynamic charge {inches}
EF	System thermal effectiveness
ER{JJ}	Broken thermocouple #
ET	System temp. effectiveness
FS	Fan speed{1150,1700,2300}
FU{I}	Enthalpy flux of 1/3 of duct upstrm evap. {KJ/s}
FD{I}	Enthalpy flux of 1/3 of duct dnstrm evap. {KJ/s}
FM{I}	1/3 of total air mass flow rate upstrm evap.{Kg/s}
FO{I}	1/3 of total air mass flow rate dnstrm evap.{Kg/s}
HE{I}	Enthalpy of air upstrm of evap. {KJ/Kg dry air}
HF{I}	Enthalpy of air dnstrm of evap. {KJ/Kg dry air}
HC{I}	Enthalpy of air upstrm of cond. {KJ/Kg dry air}
HD{I}	Enthalpy of air dnstrm of cond. {KJ/Kg dry air}
HH	Hours
HL{I}	Enthalpy of R-11 {Sat. liq.}{KJ/Kg}
HV{I}	Enthalpy of R-11 {Sat. vap.}{KJ/Kg}
H1	Average enthalpy of air upstrm of evap.{KJ/Kg dry air}
H2	Average enthalpy of air dnstrm of evap.{KJ/Kg dry air}
H3	Average enthalpy of air upstrm of cond.{KJ/Kg dry air}
H4	Average enthalpy of air dnstrm of cond.{KJ/Kg dry air}
J1	Total enthalpy of air upstrm of evap. {KJ/s}
J2	Total enthalpy of air dnstrm of evap. {KJ/s}
J3	Total enthalpy of air upstrm of cond. {KJ/s}
J4	Total enthalpy of air dnstrm of cond. {KJ/s}
JJ	Broken thermocouple counter
K	Ln of PS
LO	# of loops
LR	# of loops with R-11
M1	Total air mass flow rate upstrm evap. {Kg/s}

M2	Total air mass flow rate dnstrm evap. {Kg/s}
M3	Total air mass flow rate upstrm cond. {Kg/s}
M4	Total air mass flow rate dnstrm cond. {Kg/s}
ME{I}	1/18 of air mass flow rate at location I upstrm evap.{Kg/s}
MF{I}	1/18 of air mass flow rate at location I dnstrm evap.{Kg/s}
MC{I}	1/18 of air mass flow rate at location I upstrm cond.{Kg/s}
MD{I}	1/18 of air mass flow rate at location I dnstrm cond.{Kg/s}
MM	Minutes
MR{I}	Mass flow rate of R-11 from flowmeters {Kg/s}
MG{I}	Mass flow rate of R-11 from flowmeters {mm}
NV{I,J,K}	Normalized velocity profile,I-fan speed, J-UE/DE/UC/DC,K-1 to 18
P	Perimeter of duct {mm}
PP	Temp. scan control character
PE	Static air pressure in duct upstrm evap. {mm}
PF	Static air pressure in duct dnstrm evap. {mm}
PC	Static air pressure in duct upstrm cond. {mm}
PD	Static air pressure in duct dnstrm cond. {mm}
PS	Saturated pressure of water vapour {Pa}
Q1	Heat loss thru duct before evap.{W}
Q2	Heat loss thru duct after evap. {W}
Q3	Heat loss thru duct before cond.{W}
Q4	Heat loss thru duct after cond. {W}
Q5	Heat loss thru R-11 piping liquid side {W}
Q6	Heat loss thru R-11 piping vapour side {W}
Q7	Heat loss thru duct before furnace{W}
Q8	Heat loss thru duct after furnace {W}
QQ{I}	Average energy absorbed across 1/3 vert. strip of evap.{KJ/s}
QR{I}	Energy absorbed by evap. thru row I using mass flow rates{KJ/s}
QT	Total flux across evap.{KJ/s}
RL	Recirculation line diameter{inches}
S	Control letter for mass flow rates
SC{I}	Evap. static charge{inches}
SS	Seconds
ST	Average static charge {in.}
SU	Average static charge {%}
TA	Ambient air temp. {deg.C}
TW	Outside dew point temp. {deg.C}
TE{I}	Temp. of air before evap.{deg.C}
TF{I}	Temp. of air after evap. {deg.C}
TC{I}	Temp. of air before cond.{deg.C}
TD{I}	Temp. of air after cond. {deg.C}
TI	Estimated temp. directly before evap. {deg.C}
TJ	Estimated temp. directly after evap. {deg.C}
TK	Estimated temp. directly before cond. {deg.C}
TM	Estimated temp. directly after cond. {deg.C}
TN	Estimated temp. directly before furnace {deg.C}
TQ	Estimated temp. directly after furnace {deg.C}
TL{I}	Temp. of R-11 at liquid side of evap.{deg.C}
TV{I}	Temp. of R-11 at vapour side of evap.{deg.C}
TP	Average R-11 liquid line temp.{deg.C}
TB	Average R-11 vapour line temp.{deg.C}
T1	Average air temp. before evap.{deg.C}
T2	Average air temp. after evap. {deg.C}

T3	Average air temp. before cond.{deg.C}
T4	Average air temp. after cond. {deg.C}
T9	Maximum delta air temp in duct {deg.C}
UU	If 1, then printing to paper
U1	Conductance thru duct before evap.{m^2hc/W}
U2	Conductance thru duct after evap. {m^2hc/W}
U3	Conductance thru duct before cond.{m^2hc/W}
U4	Conductance thru duct after cond. {m^2hc/W}
V1	Average air velocity before evap.{m/s}
V2	Average air velocity after evap. {m/s}
V3	Average air velocity before cond.{m/s}
V4	Average air velocity after cond. {m/s}
VR{1}	Upstrm evap. reference air velocity {FPM}
VR{2}	Dnstrm evap. reference air velocity {FPM}
VR{3}	Upstrm cond. reference air velocity {FPM}
VR{4}	Dnstrm cond. reference aie velocity {FPM}
VE{I}	Velocity at I of air upstrm evap.{mpm}
VF{I}	Velocity at I of air dnstrm evap.{mpm}
VC{I}	Velocity at I of air upstrm cond.{mpm}
VD{I}	Velocity at I of air dnstrm cond.{mpm}
VT{I}	Corrected reference velometer reading {m/s}
VU{I}	Corrected reference velometer reading {fpm}
W	Humidity ratio
Y1	Coeff. used for duct heat loss upstrm evap.
Y2	Coeff. used for duct heat loss dnstrm evap.
Y3	Coeff. used for duct heat loss upstrm cond.
Y4	Coeff. used for duct heat loss dnstrm cond.
Y7	Coeff. used for duct heat loss upstrm furnace
Y8	Coeff. used for duct heat loss dnstrm furnace
YY	Control variable used for correction of mass flow rates
Z	Control variable for correction of input data

STRING VARIABLES:

A$	
K1$	Strings used in temperature
KH$	scan routine
KK$	
CN$	
AB$	If AB,then abort loading data from disk
CS$	Y-add comments to disk data
DA$	Date
D$	CHR${4}
E$	CHR${13} {carrage return}
EE$	If Y, use data saved on disk from last test
MM$	Used in main menu to channel selection
NN$	Test name {reading file from disk}
QQ$	
QY$	Strings used to store comments of run
QZ$	
RD${I}	Recirculation line descriptors{NF-no flow/LF-low flow/F-flow/ SF-slug flow}
S$	Used in correction of mass flow rates/to abort print/to return t main menu after print
T$	Test #
U$	Control string variable for calling stored data on disk

U1$	Control var.used in recalling test 7 or 5a from disk{Y or N}
U2$	Control var.used in recalling tests previous to test 7{Y or N}
VV$	Test name {saving file to disk}
WW$	Glass or Brass float {G ,B}
X$	****************
Z$	Misc. string variable{1060,1910}

DATA ACQUISITION PROGRAM 'TSHE' LISTING

```
10  HOME
20  PRINT : PRINT " SWITCH MONITOR TO 80 COL. CARD"
30  PRINT  CHR$ (4);"PR#3"
40  DIM CH(93),DE(18),DF(18),DC(18),DD(18),HE(18),HF(18),HC(18),HD(18),M
     E(18),MF(18),MC(18),MD(18),ER(89)
50  DIM QR(18),TE(18),TF(18),TC(18),TD(18),VE(18),VF(18),VC(18),VD(18),D
     S(5,3,4),NV(3,4,18),DV(18,4),CF(3,4)
60  DIM FM(3),FO(3),FU(3),FD(3)
70 D$ =  CHR$ (4):E$ =  CHR$ (13)
80  REM   ASSUME DAMPER#5,FAN 1700
90  FOR I = 1 TO 8:RD$(I) = "ZZ": NEXT I
100 DP = 5:FS = 1700:NR = 8:CH(89) = 999
300  FOR I = 1 TO 5: FOR J = 1 TO 3: FOR K = 1 TO 4
310  READ DS(I,J,K): NEXT K: NEXT J: NEXT I
320  DATA  .197,.774,-1.274,-.622,.228,1.471,-2.7,-1.365,.349,2.548,-4.8
     54,-2.503
330  DATA  .197,.789,-1.244,-.607,.273,1.487,-2.67,-1.32,.319,2.548,-4.8
     54,-2.442
340  DATA  .197,.774,-1.259,-.622,.258,1.487,-2.67,-1.335,.334,2.548,-4.
     824,-2.412
350  DATA  .197,.774,-1.259,-.652,.319,1.517,-2.67,-1.305,.410,2.548,-4.
     884,-2.488
360  DATA  .288,.789,-1.274,-.758,.41,1.502,-2.776,-1.578,.546,2.639,-5.
     036,-2.776
370  PRINT  CHR$ (12)
380  PRINT  TAB( 20)"   THERMOSIPHON HEAT EXCHANGER"
390  PRINT  TAB( 20)"      DATA AQUISITION PROGRAM"
400  PRINT  TAB( 36)"BY": PRINT  TAB( 32)"F.A.STAUDER": PRINT
410  INPUT " TO INPUT A TEST RUN FROM DISKETTE TYPE PT. IF NOT PRESS RET
     URN:";U$
420  IF U$ = "PT" THEN 440
430  PRINT : INPUT "FOR INITIALIZATION, SET TIME,INTERVAL SCAN(20 SEC),
     AND 'ALL DATA' ON FLUKE.                      THEN PRESS RETURN TO CO
     NTINUE";Z$: GOTO 900
440  GOSUB 500: GOSUB 1700: GOTO 1930
450  REM
500  REM  RETRIEVE PREVIOUS STORED DATA
510  PRINT D$;"CATALOG,D2"
520  PRINT : INPUT " INPUT TEST NAME:";NN$
530  PRINT : INPUT " IS THIS TEST 7 OR 5A? (Y or RETURN):";U1$
540  IF U1$ = "Y" THEN 560
550  PRINT : INPUT " IS THIS A TEST PREVIOUS  TO TEST 7? (Y or RETURN):"
     ;U2$
560  INPUT " TO ABORT LOADING PREVIOUS TEST FROM DISK, TYPE AB,    IF NOT
      PRESS RETURN:";AB$
570  IF AB$ = "AB" THEN 410
580  PRINT D$;"OPEN";NN$
590  PRINT D$;"READ";NN$
600  INPUT HH: INPUT MM: INPUT SS: INPUT DP: INPUT LO: INPUT LR: INPUT N
     R: INPUT FS: INPUT TW: INPUT PA
610  INPUT PE: INPUT PF: INPUT PC: INPUT PD: INPUT DD: INPUT T$: INPUT P
     B: INPUT TA
620  IF U2$ = "Y" THEN 660
630  FOR I = 1 TO 8
640  INPUT SC(I): INPUT ED(I): INPUT TV(I): INPUT TL(I): INPUT MG(I): INPUT
     WW$(I): INPUT RD$(I): NEXT I
650  GOTO 670
660  FOR I = 1 TO 8: INPUT SC(I): INPUT ED(I): INPUT TV(I): INPUT TL(I):
      INPUT MG(I): INPUT WW$(I): NEXT I
670  FOR I = 1 TO 18
680  INPUT TE(I): INPUT TF(I): INPUT TC(I): INPUT TD(I): NEXT I
690  INPUT M1: INPUT M2: INPUT M3: INPUT M4: INPUT E1: INPUT E2: INPUT E
     3: INPUT E4: INPUT EF: INPUT ET: INPUT QT: INPUT QQ$: INPUT DA$: INPUT
     QZ$: INPUT QY$
700  INPUT VR(1): INPUT VR(2): INPUT VR(3): INPUT VR(4)
710  IF U1$ = "Y" OR U2$ = "Y" THEN 730
720  INPUT CH(89)
730  PRINT D$;"CLOSE";NN$,D2
740  IF FS = 1150 THEN FF = 1
750  IF FS = 1700 THEN FF = 2
760  IF FS = 2300 THEN FF = 3
770  RETURN
780  REM
900  PRINT D$;"BLOAD FLUKES.BJO,D1"
910  PRINT D$;"IN#2": INPUT "";A$: PRINT D$;"IN#0"
920  PRINT : PRINT : PRINT "        SELECT ONE OF THE FOLLOWING": PRINT
930  PRINT "RUN EXPERIMENT .......................PRESS RETURN"
940  PRINT "RUN EXPER. WITHOUT IDENT. INFO........TYPE S,RETURN"
950  PRINT "SCAN TEMPERATURES.....................TYPE T,RETURN"
960  INPUT S$: IF S$ = "T" THEN MM$ = "T": GOTO 3100
1000  PRINT  CHR$ (12);"INPUT THE FOLLOWING INFORMATION"
1010  PRINT : PRINT
1020  IF S$ = "S" THEN 1300
1030  INPUT "TEST NUMBER:";T$: IF Z <  > 0 THEN 1620
1040  INPUT "USE DESCRIPTION DATA STORED ON DISK FROM LAST RUN?(Y or RET
      URN):";EE$
1050  IF EE$ <  > "Y" THEN 1120
1055  REM
1060  PRINT D$;"OPEN LASTTEST,D1"
1070  PRINT D$;"READ LASTTEST"
1080  FOR II = 1 TO 8: INPUT SC(II): NEXT II
1090  INPUT DA$: INPUT DP: INPUT LO: INPUT LR: INPUT NR: INPUT FS: INPUT
      PB: INPUT PE$: INPUT PF$: INPUT PC$: INPUT PD$
1100  PRINT D$;"CLOSE"
1105  REM
```

```
1110  GOTO 1230
1120  INPUT "DAY/MONTH/YEAR:";DA$: IF Z <  > 0 THEN 1620
1130  INPUT "DAMPER POSITION #:";DP: IF Z <  > 0 THEN 1550
1140  INPUT "# OF LOOPS (2 OR 4):";LO: IF Z <  > 0 THEN 1620
1150  INPUT "# OF LOOPS WITH R-11:";LR: IF Z <  > 0 THEN 1620
1160  INPUT "# OF .375 IN. DIA. RECIRC. LINES :";NR: IF Z <  > 0 THEN 16
     20
1170  PRINT "INPUT THE FOLLOWING R-11 CHARGES IN INCHES AND RECIRC.LINE
     DESCRIPTERS                (No Flow,Flow,Slug Flow)"
1180  IF Z = 0 THEN 1230
1190  PRINT : PRINT : INPUT "  ROW # TO BE CORRECTED :";I
1200  INPUT "STATIC CHARGE:";SC(I)
1210  INPUT "DYNAMIC CHARGE:";ED(I)
1220  INPUT "RECIRC. LINE  :";RD$(I): GOTO 1620
1230  PRINT : PRINT "LINE #"; TAB(-10);"STATIC"; TAB( 10);"DYNAMIC"; TAB(
     10);"RECIR.LINE"
1240  IF EE$ = "Y" THEN 1270
1250  FOR I = 1 TO 8: PRINT I; TAB( 15): INPUT SC(I): VTAB  PEEK (37): HTAB
     32: INPUT ED(I): VTAB  PEEK (37): POKE 1403,47: INPUT RD$(I)
1260  NEXT I: GOTO 1290
1270  FOR I = 1 TO 8: PRINT I; TAB( 15);SC(I): VTAB  PEEK (37): HTAB 32:
     INPUT ED(I): VTAB  PEEK (37): POKE 1403,47: INPUT RD$(I): NEXT I
1280  GOTO 1300
1290  INPUT "FAN RPM(1150,1700,2300):";FS: IF Z <  > 0 THEN 1620
1300  INPUT "DEW POINT TEMP. (DEG.F):";TW:TW = (TW - 32) / 1.8: IF Z <  >
     0 THEN 1620
1310 TW =  INT (TW * 100 + .5) / 100
1320  PRINT : PRINT "INPUT THE FOLLOWING REFERENCE VELOMETER READINGS(FP
     M)"
1330  INPUT "UPSTREAM EVAP.(#27)=";VR(1)
1340  INPUT "DNSTREAM EVAP.(#07)=";VR(2)
1350  INPUT "UPSTREAM COND.(#67)=";VR(3)
1360  INPUT "DNSTREAM COND.(#50)=";VR(4): PRINT : IF Z <  > 0 THEN 1620
1370  IF EE$ = "Y" THEN 1440
1380  INPUT "INPUT ROOM BAROMETRIC PRESSURE IN in. OF HG.:";PB: IF Z <  >
     0 THEN 1620
1390  PRINT : PRINT "INPUT THE FOLLOWING STATIC PRESSURES IN MM. OF HG.
     . IF NOT AVAILABLE TYPE D FOR DEFAULT VALUES"
1400  INPUT "UPSTREAM EVAP.(#27)=";PE$
1410  INPUT "DNSTREAM EVAP.(#07)=";PF$
1420  INPUT "UPSTREAM COND.(#67)=";PC$
1430  INPUT "DNSTREAM COND.(#50)=";PD$
1440  IF FS = 1150 THEN FF = 1
1450  IF FS = 1700 THEN FF = 2
1460  IF FS = 2300 THEN FF = 3
1470  IF PE$ = "D" THEN PE = DS(DP,FF,1): GOTO 1490
1480 PE =  VAL (PE$)
1490  IF PF$ = "D" THEN PF = DS(DP,FF,2): GOTO 1510
1500 PF =  VAL (PF$)
1510  IF PC$ = "D" THEN PC = DS(DP,FF,3): GOTO 1530
1520 PC =  VAL (PC$)
1530  IF PD$ = "D" THEN PD = DS(DP,FF,4): GOTO 1550
1540 PD =  VAL (PD$)
1550  IF LR = 0 THEN LR = 4:LO = 4
1560  IF DP = 5 THEN DZ = 0: GOTO 1610
1570  IF DP = 4 THEN DZ = 15: GOTO 1610
1580  IF DP = 3 THEN DZ = 30: GOTO 1610
1590  IF DP = 2 THEN DZ = 50: GOTO 1610
1600  IF DP = 1 THEN DZ = 75
1610 PE =  INT (PE * 100 + .5) / 100:PF =  INT (PF * 100 + .5) / 100:PC =
      INT (PC * 100 + .5) / 100:PD =  INT (PD * 100 + .5) / 100
1620 Z = 0: PRINT  CHR$ (12)
1630  GOSUB 1700: GOTO 1930
1690  REM
1695  REM   PRINT DATA SUBROUTINE
1700  PRINT  TAB( 9);"1. TEST #";T$
1710  PRINT  TAB( 9);"2. DATE:";DA$
1720  PRINT  TAB( 9);"3. DAMPER POSITION (DEG.OPEN).....";DZ
1730  PRINT  TAB( 9);"4. # OF LOOPS:....................";LO
1740  PRINT  TAB( 9);"5. # OF LOOPS WITH R-11:..........";LR
1750  PRINT  TAB( 9);"6. # OF .375 IN. RECIRC.LINES.....";NR
1760  PRINT  TAB( 9);"7. FAN RPM:.......................";FS
1770  PRINT  TAB( 9);"8. DEW PT.TEMP.(DEG.C):...........";TW
1780  PRINT  TAB( 9);"9. REFER.AIR VELO.(FPM)"
1790  PRINT  TAB( 13);"UPSTREAM EVAP.:";VR(1);"     DNSTREAM EVAP.:";VR(
     2)
1800  PRINT  TAB( 13);"UPSTREAM COND.:";VR(3);"     DNSTREAM COND.:";VR(
     4)
1810  PRINT  TAB( 9);"10.EVAP. R-11 CHARGES(IN.)"
1820  PRINT "ROW#       1     2     3     4     5     6     7     8"
1830  PRINT "STATIC";: HTAB 10: PRINT SC(1);: HTAB 17: PRINT SC(2);: HTAB
     23: PRINT SC(3);: HTAB 29: PRINT SC(4);: HTAB 35: PRINT SC(5);
1840  POKE 1403,39: PRINT " ";SC(6);: POKE 1403,45: PRINT " ";SC(7);: POKE
     1403,51: PRINT " ";SC(8)
1850  PRINT "DYNAM.";: HTAB 10: PRINT ED(1);: HTAB 17: PRINT ED(2);: HTAB
     23: PRINT ED(3);: HTAB 29: PRINT ED(4);: HTAB 35: PRINT ED(5);
1860  POKE 1403,39: PRINT " ";ED(6);: POKE 1403,45: PRINT " ";ED(7);: POKE
     1403,51: PRINT " ";ED(8)
1870  PRINT "REC.L.";: HTAB 10: PRINT RD$(1);: HTAB 17: PRINT RD$(2);: HTAB
     23: PRINT RD$(3);: HTAB 29: PRINT RD$(4);: HTAB 35: PRINT RD$(5);
1880  POKE 1403,39: PRINT "     ";RD$(6);"     ";RD$(7);"     ";RD$(8)
1890  PRINT  TAB( 9);"11.DUCT STATIC PRESSURES(MM OF HG)"
1900  PRINT  TAB( 13);"UPSTREAM EVAP.=";PE;: HTAB 38: PRINT "DNSTREAM EV
     AP.=";PF
1910  PRINT  TAB( 13);"UPSTREAM COND.=";PC;: HTAB 38: PRINT "DNSTREAM CO
     ND.=";PD
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1920  PRINT  TAB( 9);"12.ROOM BAROMETRIC PRESSURE(in.OF HG)..";PB: RETURN

1925  REM
1930  PRINT : INPUT " IF DATA ABOVE CORRECT PRESS RETURN. IF NOT TYPE TH
     E # CORRESPONDING TO THE         INCORRECT DATA: ";Z$
1940  IF Z$ = "" THEN 2000
1950 Z =  VAL (Z$): PRINT  CHR$ (12)
1960  ON Z GOTO 1030,1120,1130,1140,1150,1160,1290,1300,1320,1170,1390,1
     380
1995  REM
2000  REM    VELOCITY PROFILES
2010  FOR II = 1 TO 3: FOR I = 1 TO 4: FOR J = 1 TO 18: READ NV(II,I,J)
2020  NEXT J: NEXT I: NEXT II
2030  DATA  1,1.0537,.8641,1,.9463,1.025,1.0894,1,1.0107,.9642,1,.9785,.
     8927,.9821,.8641,1,1.1252,1.0429
2040  DATA  1.0156,1.0626,1.025,.9687,.9531,.9343,1.0782,1,.9343,.9218,.
     9218,.8404,.8905,.8905,.8122,.5619,.6401,.7809
2050  DATA  .7165,.7732,1,1.0794,1.0378,.9811,.9887,1,.9811,1.0643,1,1,.
     9887,.9433,.9811,.9282,1,.9433
2060  DATA  .7402,.9687,.903,.6777,1,1.0438,.7528,.9844,1.0595,.8279,1,.
     975,.7402,.9531,1.072,.7528,.9218,.9186
2070  DATA  .9935,1.0762,.8673,.9434,.9456,.963,.9935,1,.9521,.8237,1.06
     53,.9804,.815,.9674,.876,.9238,1.0544,1
2080  DATA  1.0306,1.0715,.9939,1.0409,1.0102,.9469,1.0306,1,.9285,.9898
     ,.9591,.8141,.9489,.9183,.8039,.5608,.6629,.7957
2090  DATA  .9825,1.0375,1.025,1.2953,1.0876,.97,1.1827,1,.9349,1.428,.9
     875,.9349,1.05,.8874,.8698,1.2728,1.0125,.8573
2100  DATA  .6928,.9604,.9663,.667,.9802,.998,.8077,.9604,.9663,.8494,1,
     .9564,.7245,.9405,.9564,.6591,.8613,.8236
2110  DATA  .937,.9928,.927,.8382,.8583,.9599,.9313,1,.947,.811,.9714,.9
     599,.7838,.9284,.9027,.8382,.9571,.9728
2120  DATA  .9929,1.0286,1.0515,1.0214,.9714,.9242,1.0715,1,.9814,1.0071
     ,.9571,.8399,.9643,.95,.8542,.5926,.7141,.8399
2130  DATA  1.1018,.9539,.887,1.062,1,.9014,1.1861,1,.93,1.2243,.9539,.8
     664,1.0398,.8759,.8441,.9936,.938,.8298
2140  DATA  .7061,.93,.9902,.5969,.993,1.0504,.7495,.944,1.0014,.7495,1,
     1.0868,.6781,.944,1.0084,.5969,.93,.8796
2150  REM  INPUT OF VEL.PROFILE CORRECTION FACTOR
2160  FOR I = 1 TO 3: FOR II = 1 TO 4: READ CF(I,II): NEXT II: NEXT I
2170  FOR I = 1 TO 4:VU(I) = VR(I) * CF(FF,I): NEXT I
2180  DATA  .908,.961,1.179,.999
2190  DATA  .948,.973,1.050,1.036
2200  DATA  .974,.944,1.055,1.036
2300  REM       VELOMETER CALIBRATION
2310  FOR I = 1 TO 4:VT(I) = .262827199 + 4.81968389E - 3 * VU(I): NEXT
     I
2320  FOR I = 1 TO 18:VE(I) = VT(1) * NV(FF,1,I):VF(I) = VT(2) * NV(FF,2
     ,I)
2330 VC(I) = VT(3) * NV(FF,3,I):VD(I) = VT(4) * NV(FF,4,I): NEXT I
2400  REM      DAMPER POSITION PROFILE
2410  IF FF <  > 3 THEN 2600
2420  IF DP = 5 THEN 2600
2430  FOR I = 1 TO 18: FOR II = 1 TO 4: READ DV(I,II): NEXT II
2440 VC(I) = VC(I) * DV(I,DP): NEXT I
2450  DATA  1.016,1.016,1.016,1.008,1.027,1.035,1.035,1.018,1.008,1.024,
     1.024,1.016
2460  DATA  1.042,1.035,1.021,1.021,1.024,1.024,1.024,1.024,1.045,1.045,
     1.038,1.038
2470  DATA  .994,.994,.988,.988,1.024,1.024,1.032,1.032,1.016,1.024,1.02
     4,1.024
2480  DATA  1.012,1.019,1.019,1.019,1.026,1.026,1.026,1.017,1.025,1.033,
     1.033,1.025
2490  DATA  1.024,1.024,1.024,1.024,1.064,1.064,1.064,1.064,1.025,1.025,
     1.025,1.017
2500  DATA  1.053,1.023,1.023,1.015,1.042,1.042,1.042,1.033,1.026,1.026,
     1.017,1.017
2595  REM
2600  PRINT : PRINT "  SAVING RUN DESCRIPTION DATA": PRINT
2610  PRINT D$;"OPEN  LASTTEST,D1"
2620  PRINT D$;"WRITE  LASTTEST"
2630  FOR II = 1 TO 8: PRINT SC(II): NEXT II
2635  PRINT DA$;E$;DP;E$;LO;E$;LR;E$;NR;E$;FS;E$;PB;E$;PE$;E$;PF$;E$;PC$
     ;E$;PD$
2640  PRINT D$;"CLOSE"
2690  REM
3000  REM    MAIN MENU
3010  PRINT  CHR$ (12);PP = 0
3020  PRINT : PRINT "  MAIN MENU ": PRINT
3030  PRINT " TO RUN EXPERIMENT..............................TYPE R"
3040  PRINT " TO SCAN TEMPERATURES...........................TYPE T"
3050  PRINT " TO CHANGE INPUT DATA...........................TYPE C"
3060  PRINT " TO CALC. RUN USING DISK DATA...................TYPE D"
3070  PRINT " TO INPUT PREVIOUS RUN DATA SAVED ON DISK......TYPE P"
3080  INPUT "       SELECTION?  :";MM$
3090  IF MM$ = "R" THEN  GOSUB 3150: GOTO 4000
3100  IF MM$ = "T" THEN PP = 1: GOSUB 3150: GOSUB 6000: GOTO 3000
3110  IF MM$ = "C" THEN 1630
3120  IF MM$ = "P" THEN  CLEAR : GOTO 40
3130  IF MM$ = "D" THEN  GOSUB 3460: GOTO 4130
3140  GOTO 3000
3145  REM
3150  REM   SCAN OF TEMPERATURES
3160  PRINT : INPUT "SET FLUKE TO SCAN FROM CHANNEL 0 TO 89, THEN PRESS
     RETURN TO CONTINUE:";S$: PRINT
3170  IF PP = 1 THEN 3190
3180  PRINT "  OBTAIN R-11 MASS FLOW RATES WHILE SCANNING TEMPERATURES "
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FLUME INTERFACE PROGRAM

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3190  PRINT : PRINT  TAB( 25);"ONE MOMENT PLEASE"
3200 D$ =  CHR$ (4)
3210 C$ = "01234567890123456789O": POKE 254, PEEK (131): POKE 255, PEEK
     (132):K% =  PEEK (255) * 256 +  PEEK (254) + 1:K1% = K% + 1
3220  CALL 768
3230 KK = 32768
3240 A =  PEEK (KK): IF A = 65 THEN 3220
3250  IF A <  > 89 THEN KK = KK + 1: GOTO 3240
3260 KH% = KK / 256: POKE K1%,KH%: POKE K%,KK - KH% * 256:DD =  VAL ( MID$
     (C$,2,3)):HH =  VAL ( MID$ (C$,6,2)):MM =  VAL ( MID$ (C$,9,2)):SS =
      VAL ( MID$ (C$,12,2))
3270  PRINT DD;" ";HH;" ";MM;" ";SS
3280 KK = KK + 22
3290 KK = KK + 1: IF  PEEK (KK) <  > 65 GOTO 3290
3300 KH% = KK / 256: POKE K1%,KH%: POKE K%,KK - KH% * 256:CN% =  VAL ( MID$
     (C$,2,3)):CH(CN%) =  VAL ( MID$ (C$,6,7)):KK = KK + 18: IF  PEEK (K
     K) = 65 GOTO 3300
3310  IF  PEEK (KK) <  > 20 THEN  PRINT "ERROR IN UPACKING,WILL SCAN AGA
                IN": GOTO 3220
3320  PRINT "          COMPLETED SCAN"
3330  FOR I = 1 TO 10: CALL  - 198: NEXT I
3340  INPUT " TO CONTINUE PRESS RETURN. TO SCAN AGAIN TYPE 'SCAN' ";A$
3350  IF A$ = "SCAN" GOTO 3220
3360  REM
3370  FOR I = 1 TO 18
3380 TE(I) = CH(19 + I):TF(I) = CH(I - 1):TC(I) = CH(59 + I):TD(I) = CH(
     39 + I): NEXT I
3390  FOR I = 1 TO 8:TV(I) = CH(80 + I): NEXT I:TA = CH(80):II = 0
3400  FOR I = 17 TO 77 STEP 20: FOR JJ = 1 TO 2:TL(JJ + II) = CH(JJ + I)
     : NEXT JJ
3410 II = II + 2: NEXT I
3420 ER = 0:JJ = 1
3430  FOR I = 0 TO 89
3440  IF  ABS (CH(I)) > 100 THEN ER(JJ) = I:JJ = JJ + 1
3450  NEXT I
3460 T1 = 0:T2 = 0:T3 = 0:T4 = 0
3470  FOR I = 1 TO 18:T1 = TE(I) / 18 + T1:T2 = TF(I) / 18 + T2:T3 = TC(
     I) / 18 + T3:T4 = TD(I) / 18 + T4: NEXT I
3480  RETURN
3490  REM
4000  REM  INPUT OF MASS FLOW RATES
4010 YY = 0
4020  PRINT  CHR$ (12): PRINT "INPUT THE FOLLOWING FLOWMETER READINGS IN
      MM.  TAKE ALL READINGS AT THE                    TOP OF EITHER BALL
     FLOAT OR LARGEST DIAMETER OF BRASS FLOAT"
4030  PRINT : PRINT "FLOW METER #   (G OR B FLOAT)    FLOW (MM)": PRINT

4040  FOR I = 1 TO 8
4050  IF YY = 1 THEN  PRINT I;: HTAB 20: PRINT WW$(I);: HTAB 37: PRINT M
     G(I): GOTO 4070
4060  PRINT I: TAB( 20): INPUT WW$(I): VTAB  PEEK (37): HTAB 37: INPUT M
     G(I)
4070  NEXT I
4080  PRINT : INPUT "TO CHANGE ANY ONE OF THE ABOVE INPUTS, TYPE IT'S CO
     RRESPONDING LINE NUMBER,                  IF OK, PRESS RETURN: ";S$
4090  IF S$ = "" THEN 4130
4100 S =  VAL (S$):YY = 1
4110  PRINT S: TAB( 20): INPUT WW$(S): VTAB  PEEK (37): HTAB 37: INPUT M
     G(S)
4120  GOTO 4020
4130  IF WW$(1) = "B" THEN MR(1) = 134.835 + 5.7521 * MG(1): GOTO 4150
4140 MR(1) = 3.53034 + 1.4064 * MG(1)
4150  IF WW$(2) = "B" THEN MR(2) = 147.156 + 5.7376 * MG(2): GOTO 4170
4160 MR(2) = 5.5607 + 1.3709 * MG(2)
4170  IF WW$(3) = "B" THEN MR(3) = 219.86 +  + 5.8469 * MG(3): GOTO 4190

4180 MR(3) = 2.7222 + 1.3988 * MG(3)
4190  IF WW$(4) = "B" THEN MR(4) = 152.354 + 5.9169 * MG(4): GOTO 4210
4200 MR(4) = 7.2705 + 1.3488 * MG(4)
4210  IF WW$(5) = "B" THEN MR(5) = 107.952 + 5.8637 * MG(5): GOTO 4230
4220 MR(5) = 2.3381 + 1.3813 * MG(5)
4230  IF WW$(6) = "B" THEN MR(6) = 131.201 + 5.8506 * MG(6): GOTO 4250
4240 MR(6) = 2.3182 + 1.3684 * MG(6)
4250  IF WW$(7) = "B" THEN MR(7) = 77.662 + 5.936 * MG(7): GOTO 4270
4260 MR(7) = 1.833 + 1.3674 * MG(7)
4270  IF WW$(8) = "B" THEN MR(8) = 105.741 + 5.8039 * MG(8): GOTO 4290
4280 MR(8) = 4.5142 + 1.3579 * MG(8)
4285  REM
4290  REM   FLOWMETER CALIBRATION DENSITY CORRECTION
4300 D5(1) = 1511.3:D5(2) = 1522.8:D5(3) = 1515.3:D5(4) = 1518.2:D5(5) =
     1513.9:D5(6) = 1534.3:D5(7) = 1516.4:D5(8) = 1511.3
4310  FOR I = 1 TO 8: IF MG(I) <  = 2 AND WW$(I) = "G" THEN MR(I) = 0
4320  IF MR(I) < 0 THEN MR(I) = 0
4330 D4 = (1935.356 - .69333 * (TL(I) + 273.15) - 2.83897E - 3 * (TL(I) +
     273.15))
4334  IF WW$(I) = "G" THEN D6 = 2530: GOTO 4340
4336 D6 = 8530
4340 MR(I) = (((D6 - D4) / (D6 - D5(I)) * D5(I) / D4) ^ .5) * MR(I) * D4
     / 6E7
4350  NEXT I
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4360  REM
5000  REM  THIS SECTION CALCULATES THE FLUXES PRESENT IN THE SYSTEM.
5010  PRINT : PRINT : PRINT  TAB( 20):"PERFORMING CALCULATIONS"
5020  PRINT  TAB( 25);"PLEASE WAIT"
5030 C1 =  - 5800.2206:C2 = 1.3914993:C3 =  - .04864024:C4 = .4176478E -
     4:C5 =  - .144521E - 7:C6 = 6.5459673:AR = 1.54838E - 2
5040 TW = TW + 273.15: REM   DEG.C  TO DEG.K
5050 K = C1 / TW + C2 + C3 * TW + C4 * TW ^ 2 + C5 * TW ^ 3 + C6 *  LOG
     (TW)
5060 PS =  EXP (K)
5070 PB = PB * 3.38638E3: REM    PB( Pa FROM in. Hg)
5080 W = .62198 * PS / (PB - PS)
5090 M1 = 0:M2 = 0:M3 = 0:M4 = 0:E1 = 0:E2 = 0:E3 = 0:E4 = 0:QT = 0:H1 =
     0:H2 = 0:H3 = 0:H4 = 0:ST = 0:J1 = 0:J2 = 0:J3 = 0:J4 = 0
5095  FOR I = 1 TO 3:FM(I) = 0:FO(I) = 0:FU(I) = 0:FD(I) = 0: NEXT I
5100 PE = PE * 133.322:PF = PF * 133.322:PC = PC * 133.322:PD = PD * 133
     .322
5110  FOR J = 1 TO 18
5120  REM   CALCULATE ENTHALPIES +DENSITIES
5130 HE(J) = TE(J) + W * (2501 + 1.805 * TE(J)):DE(J) = (PB + PE - PS) /
     (287.055 * (TE(J) + 273.15))
5140 HF(J) = TF(J) + W * (2501 + 1.805 * TF(J)):DF(J) = (PB + PF - PS) /
     (287.055 * (TF(J) + 273.15))
5150 HC(J) = TC(J) + W * (2501 + 1.805 * TC(J)):DC(J) = (PB + PC - PS) /
     (287.055 * (TC(J) + 273.15))
5160 HD(J) = TD(J) + W * (2501 + 1.805 * TD(J)):DD(J) = (PB + PD - PS) /
     (287.055 * (TD(J) + 273.15))
5170  REM   CALCULATE MASS FLOW RATE + FLUX
5180 ME(J) = DE(J) * VE(J) * AR:M1 = ME(J) + M1:J1 = HE(J) * ME(J) + J1
5190 MF(J) = DF(J) * VF(J) * AR:M2 = MF(J) + M2:J2 = HF(J) * MF(J) + J2
5200 MC(J) = DC(J) * VC(J) * AR:M3 = MC(J) + M3:J3 = HC(J) * MC(J) + J3
5210 MD(J) = DD(J) * VD(J) * AR:M4 = MD(J) + M4:J4 = HD(J) * MD(J) + J4
5211  NEXT J
5213  FOR I = 1 TO 3
5214  FOR J = 0 TO 15 STEP 3
5215 FU(I) = HE(I + J) * ME(I + J) + FU(I)
5216 FD(I) = HF(I + J) * MF(I + J) + FD(I)
5217 FM(I) = ME(I + J) + FM(I):FO(I) = MF(I + J) + FO(I)
5218  NEXT J: NEXT I
5219  FOR I = 1 TO 3
5220 QQ(I) = ((FM(I) + FO(I)) / 2 * (FU(I) / FM(I) - FD(I) / FO(I))) * 1
     000
5221 QQ(I) =  INT (QQ(I) * 100 + .5) / 100: NEXT I
5225 H1 = J1 / M1:H2 = J2 / M2:H3 = J3 / M3:H4 = J4 / M4
5230 E1 = (M1 + M2) / 2 * (H1 - H2):E2 = (M3 + M4) / 2 * (H4 - H3):E3 =
     (M1 + M4) / 2 * (H1 - H4):E4 = (M1 + M3) / 2 * (H1 - H3)
5251 E1 = E1 * 1000:E2 = E2 * 1000:E3 = E3 * 1000:E4 = E4 * 1000
5260  REM   HEAT LOSS BY FLOWMETER CALCULATION
5270 B0 = 54.8322296:B1 = .701634169:B2 =  - 3.173898E - 4
5280 D0 =  - 173.48954:D1 = .631723195:D2 = 4.4477719E - 4
5290  FOR I = 1 TO 8
5310 HV(I) = (B0 + B1 * (TV(I) + 273.15) + B2 * (TV(I) + 273.15) ^ 2) *
     1000
5320 HL(I) = (D0 + D1 * (TL(I) + 273.15) + D2 * (TL(I) + 273.15) ^ 2) *
     1000
5330 QR(I) = MR(I) * (HV(I) - HL(I))
5335 QT = QR(I) + QT
5337 QR(I) =  INT (QR(I) * 100 + .5) / 100: NEXT I
5342  REM   AVERAGE VAPOUR TEMP. DIFFERENCE
5344 AC = (TV(1) + TV(2)) / 2 - (TV(3) + TV(4)) / 2
5346 AE = (TV(5) + TV(6)) / 2 - (TV(7) + TV(8)) / 2
5348 AD = (TV(3) + TV(4)) / 2 - (TV(5) + TV(6)) / 2
5349 AC =  INT (AC * 100 + .5) / 100:AD =  INT (AD * 100 + .5) / 100:AE =
      INT (AE * 100 + .5) / 100
5350  REM   HEAT LOSS CALC.(DUCT)
5360  REM   ASSUMED R-6 INSULATION ON DUCTING
5370 V1 = 0:V2 = 0:V3 = 0:V4 = 0:D1 = 0:D2 = 0:D3 = 0:D4 = 0
5380 P = 2 * (914.4 + 304.8)
5390  FOR I = 1 TO 18:V1 = VE(I) / 18 + V1:V2 = VF(I) / 18 + V2
5400 V3 = VC(I) / 18 + V3:V4 = VD(I) / 18 + V4
5410 D1 = DE(I) / 18 + D1:D2 = DF(I) / 18 + D2:D3 = DC(I) / 18 + D3:D4 =
     DD(I) / 18 + D4
5420  NEXT I
5430 U1 = (3.31 + .057 * V1) + 1 / 1.06:U2 = (3.31 + .057 * V2) + 1 / 1.
     06
5440 U3 = (3.31 + .057 * V3) + 1 / 1.06:U4 = (3.31 + .057 * V4) + 1 / 1.
     06
5445 U7 = 3.31 + .057 * V4:U8 = 3.31 + .057 * V1
5450 Y1 = 560205.33 * V1 * D1 / (U1 * P * .12)
5460 TI = (T1 * (Y1 - 1) + 2 * TA) / (Y1 + 1)
5470 Q1 = U1 * P * .12 / 1000 * ((T1 + TI) / 2 - TA)
5480 Y2 = 560205.33 * V2 * D2 / (U2 * P * .58)
5490 TJ = (T2 * (Y2 + 1) - 2 * TA) / (Y2 - 1)
5500 Q2 = U2 * P * .58 / 1000 * ((TJ + T2) / 2 - TA)
5510 Y3 = 560205.33 * V3 * D3 / (U3 * P * 1.1684)
5520 TK = (T3 * (Y3 - 1) + 2 * TA) / (Y3 + 1)
5530 Q3 = U3 * P * 1.1684 / 1000 * ((TK + T3) / 2 - TA)
5540 Y4 = 560205.33 * V4 * D4 / (U4 * P * 1.1684)
5550 TM = (T4 * (Y4 + 1) - 2 * TA) / (Y4 - 1)
5560 Q4 = U4 * P * 1.1684 / 1000 * ((TM + T4) / 2 - TA)
5562 Y7 = 173.96 * V4 * D4 / U7:TN = (T4 * (Y7 - 1) + 2 * TA) / (Y7 + 1)

5563 Q7 = U7 * 4.7915 * ((T4 + TN) / 2 - TA)
5567 Y8 = 177.697 * V1 * D1 / U8:TO = (T1 * (Y8 + 1) - 2 * TA) / (Y8 - 1
     )
```

```
5560 Q8 = Q8 * 3.1761 * ((T10 + T11) / 2 - TA)
5570  REM   HEAT LOSS THRU PIPING
5580 TP = 0:TB = 0:LL = 8 * LR / LO
5585  FOR I = 1 TO 8
5590  IF SC(I) = 0 THEN 5597
5595 TP = TL(I) / LL + TP:TB = TV(I) / LL + TB
5597  NEXT I
5600 RS = 1.1 / 2:R1 = 3 / 16:RO = .25 / 2:AF = TP * 1.8 + 32:AB = TB *
     1.8 + 32:AA = TA * 1.8 + 32:K1 = .255:K = .05048
5610 Q5 = (AP - AA) / ((RS *  LOG (R1 / RO) / K) + (RS *  LOG (RS / R1) /
     K1) + .6) * 3.14 * LR / LO * 8 * 1.1 / 12 * 19 * .2930
5620 Q6 = 1 / 6 * 1.5 * 13.75 * (AB - AA) * .2930
5630 E1 = E1 - Q2 - Q1:E2 = E2 + Q3 + Q4:E3 = E3 + Q7 + Q8:E4 = E4 - Q1 +
     Q3
5640  REM    CALCULATION OF EFFECTIVENESS
5650 EF = (E1 + E2) / 2 / E4 * 100
5660 ET = (T1 - T2) / (T1 - T3) * 100
5662 ST = 0
5663  FOR I = 1 TO 8: IF SC(I) = 0 THEN 5667
5665 ST = SC(I) / LL + ST
5667  NEXT I
5668 SU =  INT (ST / 36 * 10000 + .5) / 100
5670 TW = TW - 273.15:PB = PB / 3.3864E3:PE = PE / 133.322:PF = PF / 133
     .322:PC = PC / 133.322:PD = PD / 133.322
5672 T9 =  INT ((T1 - T3) * 100 + .5) / 100
5675  REM    ROUND OFF
5680 M1 =  INT (M1 * 1000 + .5) / 1000:M2 =  INT (M2 * 1000 + .5) / 1000
     :M3 =  INT (M3 * 1000 + .5) / 1000:M4 =  INT (M4 * 1000 + .5) / 100
     0
5685 E1 =  INT (E1 * 100 + .5) / 100:E2 =  INT (E2 * 100 + .5) / 100:E3 =
      INT (E3 * 100 + .5) / 100:E4 =  INT (E4 * 100 + .5) / 100:QT =  INT
     (QT * 100 + .5) / 100:Q5 =  INT (Q5 * 100 + .5) / 100:Q6 =  INT (Q6
     * 100 + .5) / 100
5687 EF =  INT (EF * 100 + .5) / 100:ET =  INT (ET * 100 + .5) / 100
5688  FOR I = 1 TO 8:MR(I) =  INT (MR(I) / 1E - 5 * 100 + .5) / 100 * 1E
     - 5: NEXT I
5690 T1 =  INT (T1 * 100 + .5) / 100:T2 =  INT (T2 * 100 + .5) / 100:T3 =
      INT (T3 * 100 + .5) / 100:T4 =  INT (T4 * 100 + .5) / 100
5695 W =  INT (W * 100000 + .5) / 100000:TW =  INT (TW * 100 + .5) / 100

5697 V1 =  INT (V1 * 100 + .5) / 100:V2 =  INT (V2 * 100 + .5) / 100:V3 =
      INT (V3 * 100 + .5) / 100:V4 =  INT (V4 * 100 + .5) / 100
5700  GOSUB 6000: PRINT
5710  IF MM$ = "D" THEN 5730
5720  PRINT " COMMENTS ON RUN?": INPUT QQ$,QZ$,QY$
5725  GOTO 7000
5730  INPUT "DO YOU  WISH TO ADD ANYTHING TO THE COMMENT STATEMENT? (Y o
     r RETURN):";CS$
5740  IF CS$ = "" THEN 7000
5750  PRINT : PRINT "COMMENTS:";QQ$;QZ$;QY$;: INPUT QY$
5760  GOTO 7000
5790  REM
6000  REM   PRINT ROUTINE (RESULTS)
6010  PRINT  CHR$ (12)
6020  IF JJ < 2 THEN 6070
6030 II = 1
6040  PRINT "THERMOCOUPLE  #";ER(II);"  IS OPEN!!":II = II + 1
6050  IF II < JJ THEN 6040
6060  PRINT : INPUT "TO CONTINUE LISTING, PRESS RETURN:";S$
6070  IF PP = 1 THEN 6210
6080  PRINT  CHR$ (12)
6090  PRINT : PRINT "THESE RESULTS ARE BASED ON DUCT AIR VELOCITY AND TE
     MPERATURE"
6100  PRINT : PRINT "TOTAL ENERGY ABSORBED BY EVAP.....................
     ........ ";E1;" WATTS"
6110  PRINT "TOTAL ENERGY EXPELLED BY COND.............................
      ";E2;" WATTS"
6120  PRINT "TOTAL ENERGY INPUT BY FURNACE.............................
      ";E3;" WATTS"
6130  PRINT : PRINT "ENERGY ABSORBED BY EVAP.(1/3 V.SECT. CLOSEST TO R.L
     .)...... ";QQ(3);" WATTS"
6140  PRINT "ENERGY ABSORBED BY EVAP.(MIDDLE 1/3 V.SECTION).............
      ";QQ(2);" WATTS"
6150  PRINT "ENERGY ABSORBED BY EVAP.(1/3 V.SECT.FURTHEST FROM R.L.)....
      ";QQ(1);" WATTS"
6160  PRINT : PRINT "TOTAL ENERGY ABSORBED BY EVAP.USING R-11 MASS FLOW
     RATES.. ";QT;" WATTS"
6170  PRINT : PRINT "HEAT LOST THROUGH VAPOUR SIDE OF PIPING............
     ....... ";Q6;" WATTS"
6180  PRINT "HEAT LOST THROUGH LIQUID SIDE OF PIPING...................
      ";Q5;" WATTS"
6190  PRINT : PRINT "SYSTEM THERMAL EFFECTIVENESS......................
     ....... ";EF;"%"
6200  PRINT "SYSTEM TEMP. EFFECTIVENESS................................
      ";ET;"%": PRINT
6210  PRINT "AIR FLOW RATES          MASS          VELOCITY        AVE.TEMP
     ."
6220  PRINT "                        KG/S            M/S              DEG.C"

6230  PRINT "UPSTRM EVAP.......      ";M1;: HTAB 37: PRINT V1;"
      ";T1
6240  PRINT "DNSTRM EVAP.......      ";M2;: HTAB 37: PRINT V2;"
      ";T2
6250  PRINT "UPSTRM COND.......      ";M3;: HTAB 37: PRINT V3;"
      ";T3
```

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6260  PRINT "DNSTRM COND.......     ";M4;: HTAB 37: PRINT V4;"
      ";T4
6270  IF UU = 1 THEN  PRINT  CHR$ (27); CHR$ (97); CHR$ (8): GOTO 6290
6280  PRINT " TO CONTINUE LISTING PRESS RETURN": GET S$
6290  PRINT "COND. VAPOUR HEADER(ROW #1) OUTSIDE SURFACE TEMP....  ";CH(
     89);" DEG.C"
6300  PRINT : PRINT "      TEMPERATURE DISTRIBUTIONS IN DUCT (DEG.C)"
6310  PRINT "(ALL TEMP. ARE ARRANGED LOOKING UPRIGHT,DOWNSTREAM)": PRINT

6320  PRINT "        UPSTREAM OF EVAPORATOR"
6330  FOR I = 18 TO 3 STEP  - 3
6340  HTAB 6: PRINT TE(I - 2);: HTAB 16: PRINT TE(I - 1);: HTAB 26: PRINT
     TE(I): NEXT I
6350  PRINT : PRINT "       DOWNSTREAM OF EVAPORATOR"
6360  FOR I = 18 TO 3 STEP  - 3
6370  HTAB 6: PRINT TF(I - 2);: HTAB 16: PRINT TF(I - 1);: HTAB 26: PRINT
     TF(I): NEXT I
6380  IF UU = 1 THEN 6410
6390  PRINT "TO CONTINUE LISTING TYPE RETURN";: GET S$: PRINT
6400  PRINT "TEMPERATURE DISTRIBUTION CONTINUED"
6410  PRINT : PRINT "        UPSTREAM CONDENSER"
6420  FOR I = 3 TO 1 STEP  - 1
6430  PRINT TC(I + 15);: HTAB 7: PRINT TC(I + 12);: HTAB 14: PRINT TC(I +
     9);: HTAB 21: PRINT TC(I + 6);: HTAB 28: PRINT TC(I + 3);: HTAB 35:
      PRINT TC(I)
6440  NEXT I: PRINT
6450  PRINT "         DOWNSTREAM CONDENSER"
6460  FOR I = 3 TO 1 STEP  - 1
6470  PRINT TD(I + 15);: HTAB 7: PRINT TD(I + 12);: HTAB 14: PRINT TD(I +
     9);: HTAB 21: PRINT TD(I + 6);: HTAB 28: PRINT TD(I + 3);: HTAB 35:
      PRINT TD(I)
6480  NEXT I: PRINT : PRINT
6490  IF UU = 1 THEN 6510
6500  PRINT "TO CONTINUE LISTING PRESS RETURN": GET S$
6510  PRINT "R-11     MASS FLOW    LIQ.TEMP.  VAP.TEMP.   ENERGY"
6520  PRINT "ROW#    RATES(KG/S)      DEG.C      DEG.C       KJ/S"
6530  FOR I = 1 TO 8
6540  PRINT " ";I;"          ";MR(I);: HTAB 24: PRINT TL(I);: HTAB 34: PRINT
     TV(I);: HTAB 39: PRINT "      ";QR(I): NEXT I
6550  PRINT : PRINT "AVERAGE VAPOUR TEMP. DIFFERENCE"
6560  PRINT "BETWEEN LOOPS 1 AND 2: ";AC;" DEG.C"
6570  PRINT "BETWEEN LOOPS 2 AND 3: ";AD;" DEG.C"
6580  PRINT "BETWEEN LOOPS 3 AND 4: ";AE;" DEG.C"
6590  PRINT : IF UU = 1 THEN 6620
6600  PRINT "TO INPUT COMMENTS AND CONTINUE TYPE 'C', IF NOT PRESS RETUR
     N TO SEE DATA": GET S$
6610  IF S$ <  > "C" THEN 6000
6620  RETURN
6690  REM
7000  REM   MENU #2
7010  PRINT  CHR$ (12)
7020  PRINT  TAB( 20);"MENU #2"
7030  PRINT : PRINT "TO SAVE RESULTS........................TYPE 1"
7040  PRINT "TO PRINT RESULTS ON PRINTER............TYPE 2"
7050  PRINT "TO RETURN TO MAIN MENU.................TYPE 3"
7060  PRINT "TO START AT BEGINNING..................TYPE 4"
7070  PRINT "TO END PROGRAM.........................TYPE 5"
7080  INPUT " SELECTION #? :";XX
7090  ON XX GOTO 7500,7200,3000,7100,9000
7100  PRINT  CHR$ (12): PRINT : PRINT : PRINT " TYPE  'RUN' TO RUN PROGR
     AM": CALL  - 1438
7190  REM
7200  REM  PRINTER COMMANDS
7210  PRINT : PRINT "TURN ON AND LOAD PRINTER. PRESS RETURN WHEN READY.
     "
7220  INPUT " TO ABORT PRINT, TYPE AB:";S$
7230  IF S$ = "AB" THEN 7000
7240 UU = 1: PR# 1
7250  PRINT ":"
7260  PRINT  SPC( 20);"RESULTS OF TEST # ";T$: PRINT
7270 X$ = "*****************************************************************
     *********"
7280  PRINT X$: PRINT
7290  PRINT  TAB( 9);"1. DATE:";DA$
7300  PRINT  TAB( 9);"2. TIME:";HH":";MM":";SS
7310  GOSUB 1710
7320  PRINT  TAB( 9)"13.CONDENSER INCLINATION...............45 DEGREES"
7330  PRINT  TAB( 9)"14.HUMIDITY RATIO......................";W
7340  PRINT  TAB( 9)"15.ROOM TEMPERATURE...................";TA;" DEG.C
     "
7350  PRINT  TAB( 9)"16.DELTA TEMPERATURE..................";T9;" DEG.C
     "
7360  PRINT  TAB( 9)"17.AVERAGE STATIC CHARGE...............";SU;" %"
7370  PRINT : PRINT X$
7380  GOSUB 6090
```

```
7390  PRINT X$
7400 UU = 0: PRINT : PRINT "COMMENTS": PRINT
7410  PRINT QQ$;QZ$;QY$
7420  PRINT : PRINT X$: PRINT : PRINT : PRINT : PRINT : PRINT : PRINT : PRINT
     : PRINT : PRINT : PR# 3: GOTO 7000
7495  REM
7500  REM  SAVE RESULTS
7510  PRINT  CHR$ (12):E$ =  CHR$ (13)
7520  PRINT : PRINT "  PLACE DATA DISK IN DRIVE 2": PRINT
7530  INPUT " TEST NAME?:";VV$
7540  PRINT : INPUT "TO ABORT SAVE ROUTINE TYPE AB, IF NOT TYPE RETURN";
     AB$
7550  IF AB$ = "AB" THEN 7710
7560 D$ =  CHR$ (4)
7570  PRINT D$;"OPEN";VV$",D2"
7580  PRINT D$;"WRITE";VV$
7590  PRINT HH;E$;MM;E$;SS;E$;DP;E$;LO;E$;LR;E$;NR;E$;FS;E$;TW;E$;PA
7600  PRINT PE;E$;PF;E$;PC;E$;PD;E$;DD;E$;T$;E$;PB;E$;TA
7610  FOR I = 1 TO 8
7620  PRINT SC(I);E$;ED(I);E$;TV(I);E$;TL(I);E$;MG(I);E$;WW$(I);E$;RD$(I
     )
7630  NEXT I
7640  FOR I = 1 TO 18
7650  PRINT TE(I);E$;TF(I);E$;TC(I);E$;TD(I)
7660  NEXT I
7670  PRINT M1;E$;M2;E$;M3;E$;M4;E$;E1;E$;E2;E$;E3;E$;E4;E$;EF;E$;ET;E$;
     QT;E$;QQ$;E$;DA$;E$;QZ$;E$;QY$
7680  PRINT VR(1);E$;VR(2);E$;VR(3);E$;VR(4)
7690  PRINT CH(89)
7700  PRINT D$;"CLOSE"
7710  GOTO 7000
9000  PRINT  CHR$ (12)
9010  PRINT "   END OF PROGRAM"
```

PURGE PROGRAM LISTING

FLUKE INTERFACE PROGRAM

```
]LIST

10  HOME : PRINT "THIS PROGRAM REQUIRES USE OF 80 COLUMN CARD"
20  PRINT  CHR$ (4);"PR#3"
30  DIM CH(90)
40 D$ =  CHR$ (4)
50 L$ = "OK TO PURGE"
60  PRINT  CHR$ (12)
70  HTAB 13: PRINT "THERMOSIPHON HEAT EXCHANGER"
80  HTAB 19: PRINT "PURGING PROGRAM"
90  HTAB 20: PRINT "BY F.A.STAUDER"
100  PRINT : PRINT
110  INPUT "INPUT ROOM BAROMETRIC PRESSURE IN in. OF Hg. :";PB
120 PM = 3.38638E3 * PB / 1E6 + .009
130  REM   SCAN OF TEMPERATURES
140  PRINT : PRINT : INPUT "SET FLUKE TO SCAN FROM CHANNNEL 0 TO 89, 30
     SECOND INTERVAL, AND EXTERNAL ENABLE    ALL DATA, THEN PRESS RETURN
      TO CONTINUE: ";S$: PRINT
150  PRINT "            INITIALIZING DATA LOGGER"
160  PRINT "                  PLEASE WAIT"
170  PRINT D$;"BLOAD FLUKES.BJO,D1"
180  PRINT D$;"IN#2": INPUT "";A$: PRINT D$;"IN#0"
190 C$ = "01234567890123456789O": POKE 254, PEEK (131): POKE 255, PEEK (
     132):K% =  PEEK (255) * 256 +  PEEK (254) + 1:K1% = K% + 1
200  CALL 768
210 KK = 32768
220 A =  PEEK (KK): IF A = 65 THEN 200
230  IF A <  > 89 THEN KK = KK + 1: GOTO 220
240 KH% = KK / 256: POKE K1%,KH%: POKE K%,KK - KH% * 256:DD =  VAL ( MID$
     (C$,2,3)):HH =  VAL ( MID$ (C$,6,2)):MM =  VAL ( MID$ (C$,9,2)):SS =
      VAL ( MID$ (C$,12,2))
250 KK = KK + 22
260 KK = KK + 1: IF  PEEK (KK) <  > 65 GOTO 260
270 KH% = KK / 256: POKE K1%,KH%: POKE K%,KK - KH% * 256:CN% =  VAL ( MID$
     (C$,2,3)):CH(CN%) =  VAL ( MID$ (C$,6,7)):KK = KK + 18: IF  PEEK (K
     K) = 65 GOTO 270
280  IF  PEEK (KK) <  > 20 THEN  PRINT "ERROR IN UPACKING,WILL SCAN AGA
               IN": GOTO 200
290  PRINT
300  PRINT "         COMPLETED SCAN"
310  FOR I = 1 TO 10: CALL  - 198: NEXT I
320  PRINT "TIME OF SCAN: ";HH;":";MM;":";SS
330  PRINT "ROW #  V.TEMP.   L.TEMP.   CONDITION"
340  FOR I = 1 TO 8
350 II = 0
360 L$ = "OK TO PURGE"
370  FOR IJ = 17 TO 77 STEP 20: FOR JJ = 1 TO 2:TL(JJ + II) = CH(JJ + IJ
     ): NEXT JJ:II = II + 2: NEXT IJ
380 T(1) = TL(I):T(2) = CH(80 + I)
390  FOR J = 1 TO 2
400 P(J) =  - 4.46315289 + .0540604442 * (T(J) + 273.15) - 2.20978574E -
     4 * (T(J) + 273.15) ^ 2 + 3.05397038E - 7 * (T(J) + 273.15) ^ 3
410  NEXT J
420  IF P(1) < PM OR P(2) < PM THEN L$ = "DO NOT PURGE!!!!"
430  PRINT " ";I; TAB( 6);CH(80 + I);: HTAB 18: PRINT TL(I);"       ";L
     $
440  IF L$ = "DO NOT PURGE!!!!" THEN 460
450  GOTO 470
460  FOR J = 1 TO 3: CALL  - 198: NEXT J
470  NEXT I
480  PRINT : PRINT "SCANNING TEMPERATURES AGAIN"
490  PRINT "TO EXIT SCAN,TAKE OFF EXTERNAL ENABLE ALL DATA, AND PRESS (C
     TRL) RESET"
500  PRINT " THEN TYPE   PR#3, (RETURN)
510  PRINT : PRINT
520  GOTO 190
530  END
```

APPENDIX H

ERROR ANALYSIS

UNCERTAINTY ANALYSIS:

The Kline and McKlintok {14} method of uncertainty analysis was used in this study to determine the expected magnitude of error involved with each of the calculated performance parameters. The Kline and McKlintok method is essentially,

$$Ws = \left[\left(\frac{\partial S}{\partial N1} W1 \right)^2 + \left(\frac{\partial S}{\partial N2} W2 \right)^2 + \ldots\ldots + \left(\frac{\partial S}{\partial Nn} Wn \right)^2 \right]^{1/2}$$

Where : S is a function of N1,N2,N3....

Ws is the calculated expected error associated with S

Wn is the error associated with Nn

Table H.1 summarizes the variables used, their magnitude, and the estimated or calculated error associated with each for test 'E' of test sequence 1.

TABLE H.1
ERROR ANALYSIS SUMMARY

VARIABLE	TYPICAL VALUE	PROBABLE ERROR	CALCULATED VALUE *
TEMPERATURE {from data logger}	45.1 C	+-2 C	-----
DEW POINT TEMP	61.0 C	+-3 F	-----
BAROMETRIC PRESSURE	29.564 in.Hg	+-.01	-----
REFERENCE VELOMETER READINGS	UE=415 FPM	+-10 FPM	-----
DUCT STATIC PRESSURE	50 MM MAN. FLUID	+- .5 MM	-----
R-11 FLOWMETERS	B 50 MM	+- 2.5 MM	-----
EVAP. CHARGE			
STATIC	30 in. R-11	+- .0625 in.	-----
DYNAMIC	15 iN. R-11	+- .0625 in.	-----
HUMIDITY RATIO	.0116	+-.0004	*
AIR MASS FLOW RATE	DE=.596 Kg/s	+-.010 Kg/s	*
AIR VELOCITY	DE=2.03 m/s	+- 0.05 m/s	*
AVERAGE AIR TEMP.	DE=51.5 C	+- 2.0 C	*
R-11 MASS FLOW RATE	.0203 Kg/s	+-.001 Kg/s	*
ENERGY ABSORBED IN EACH ROW	3432 W	+-170 W	*
TOTAL ENERGY ABSORBED BY EVAP. USING FLOWMTR DATA	19360 W	+-884 W	*
TOTAL ENERGY ABSORBED BY EVAP.	16170 W	+-900 W	*
TOTAL ENERGY ABSORBED BY COND.	15899 W	+-740 W	*
TOTAL ENERGY INPUT BY FURNACE	15374 W	+-910 W	*
SYSTEM THERMAL EFFECTIVENESS (%)	54	+-2.5 {+-5%}	*

APPENDIX I

LOOP PREDICTION ANALYSIS

LOOP PREDICTION ANALYSIS

The following analysis was used to predict the performance of 2, 3, and 4 loop TSHE systems using curvefitted single loop experimental data. A listing of the Apple Basic program and the curvefitted single loop equations can be found at the end of this Appendix.

PERFORMANCE PREDICTION EQUATIONS:

Refer to Figure I.1 for nomenclature associated with this analysis.

Single Loop: use loop #1

- assume an overall temperature difference, say:

E1 = 40 DEGREES C
C1 = 10 DEGREES C

THEREFORE: C4 = 10 C

overall temperature difference(OTD): OTD1 = 40 - 10 = 30 C

FROM EXPERIMENTAL DATA; EFF. = 22.75%

$$\text{SINCE EFF.} = \frac{\text{TEMP. DROP ACROSS HEAT EXCHANGER}}{\text{OTD}}$$

THEREFORE: TEMP. DROP ACROSS LOOP#1 IS 22.75 (30) /100 = 6.83

: DT1 = 6.83 C

THEREFORE: E2 = E1 - DT1
= 40 - 6.83 = 33.17 C

C5 = C4 + DT1
= 10 + 6.83 = 16.83 C

2 LOOPS: use loops # 1 and 2

- C3 = 10
  E2=33.17

- THEREFORE OTD2 = 33.17 - 10 = 23.17 C

EFF. = 19.75%

DT2 = 19.75 (23.17)/100 = 4.58 C

- C4 = C3 + 4.58
  = 10 + 4.58 = 14.58 C

THEREFORE: OTD1 = 40 - 14.58 = 25.58 C
THEREFORE RECALULATE EFF. FOR LOOP 1: EFF. = 21.25%

- DT1 = 5.40 C

- E2 = 40 - 5.40 = 34.6 C

OTD2 = 34.60 - 10 = 24.60 C ....

ITERATION CONTINUES UNTIL VERY LITTLE CHANGE OCCURS IN THE EFFECTIVENESS OF ALL THE LOOPS FOR SMALL TEMPERATURE INCREMENTS. THIS SAME METHOD WAS APPLIED TO 3 AND 4 LOOP TSHE CONFIGURATIONS.

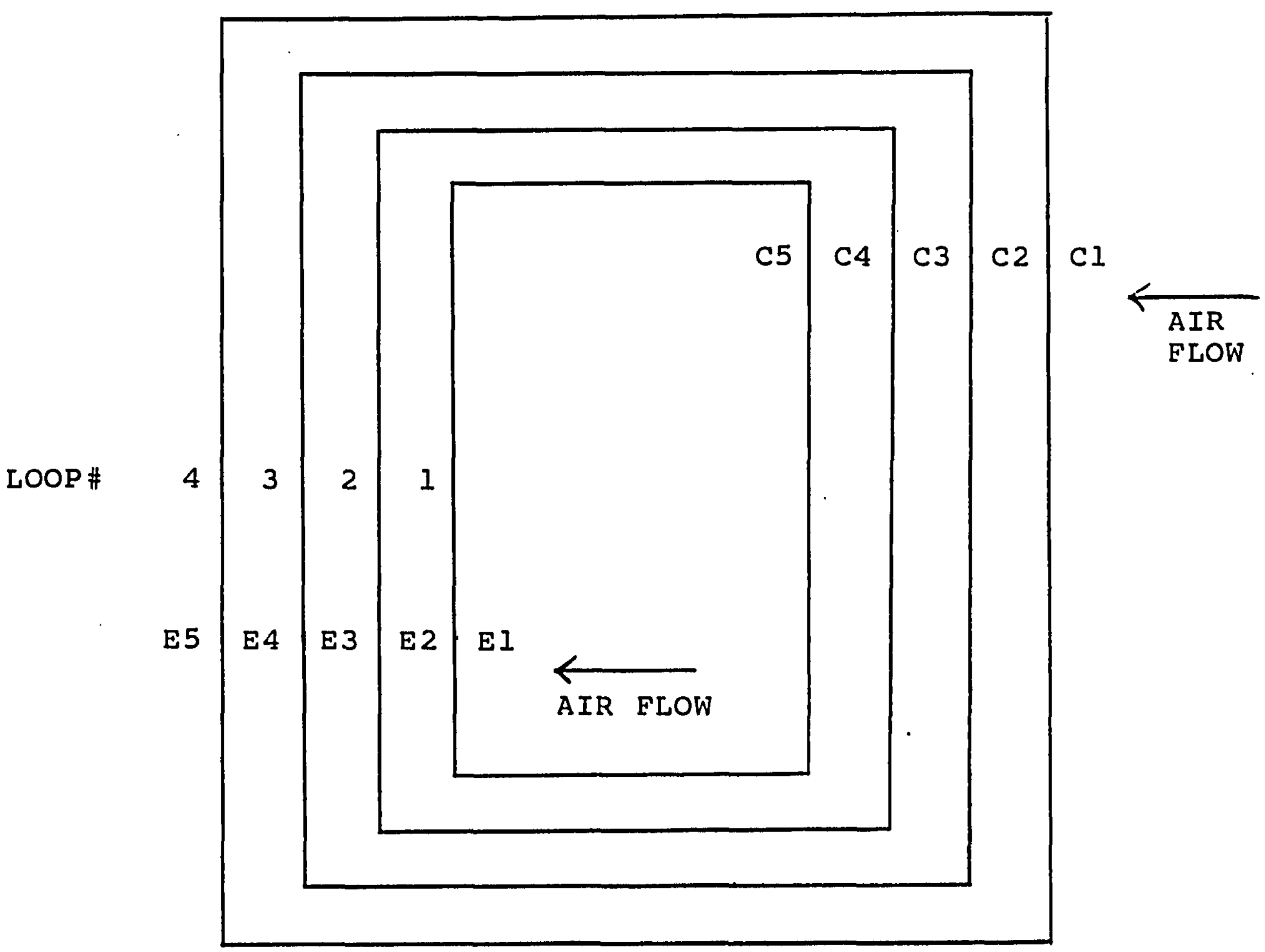
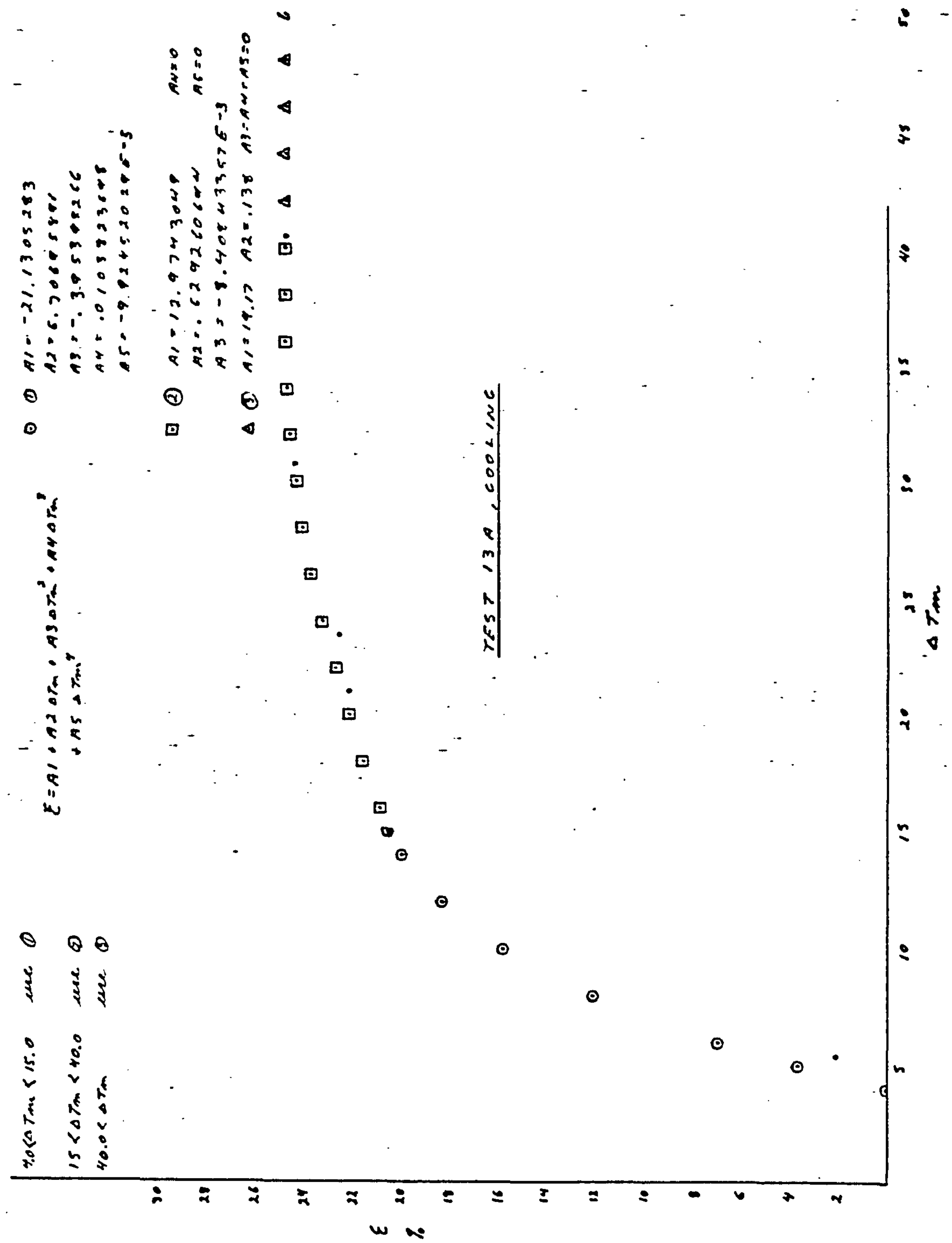


FIGURE I.1
VARIABLE NAMES USED IN MULTI-LOOP PREDICTED PERFORMANCE PROGRAM


4.0<ΔTm<15.0 use ①
15<ΔTm<40.0 use ②
40.0<ΔTm use ③
A1=-21.1305283
A1=14.17 A2=.138 A3=A4=A5=0
A4=0
A5=0
TEST 13A, COOLING
ΔTm
E %

TEST13A COOLING

Y=A+BX+CX^2+DX^3+EX^4
WHERE A=-21.1305283
B=6.70695891
C=-.395398266
D=.0103823698
E=-9.92452029E-05

X	Y
14	19.9454553
14.1	20.0098322
14.2	20.0727168
14.3	20.1341379
14.4	20.1941236
14.5	20.2527022
14.6	20.3099015
14.7	20.3657492
14.8	20.4202726
14.9	20.473499
15	20.5254552
15.1	20.5761679
15.2	20.6256636
15.3	20.6739685
15.4	20.7211085
15.5	20.7671095
15.6	20.8119968
15.7	20.8557957
15.8	20.8985313
15.9	20.9402283

TEST13 COOLING

Y=A+BX+CX^2+DX^3+EX^4
WHERE A=12.9743049
B=.629260694
C=-8.40843357E-03
D=0
E=0

X	Y
14	20.1359016
14.1	20.1752
14.2	20.2143302
14.3	20.2532922
14.4	20.2920861
14.5	20.3307118
14.6	20.3691693
14.7	20.4074587
14.8	20.4455799
14.9	20.4835329
15	20.5213178
15.1	20.5589344
15.2	20.596383
15.3	20.6336633
15.4	20.6707755
15.5	20.7077195
15.6	20.7444953
15.7	20.781103
15.8	20.8175425
15.9	20.8538139

TEST13A COOLING

Y=A+BX+CX^2+DX^3+EX^4
WHERE A=-21.1305283
B=6.70695891
C=-.395398266
D=.0103823698
E=-9.92452029E-05

X	Y
3	-4.29595085
3.1	-3.83859735
3.2	-3.38733516
3.3	-2.9421095
3.4	-2.5028658
3.5	-2.06954977
3.6	-1.6421073
3.7	-1.22048458
3.8	-.804628
3.9	-.394484194
4	9.99995883E-03
4.1	.408877345
4.2	.802200611
4.3	1.19002217
4.4	1.57239418
4.49999999	1.9493686
[illegible]	[illegible]

TEST13A

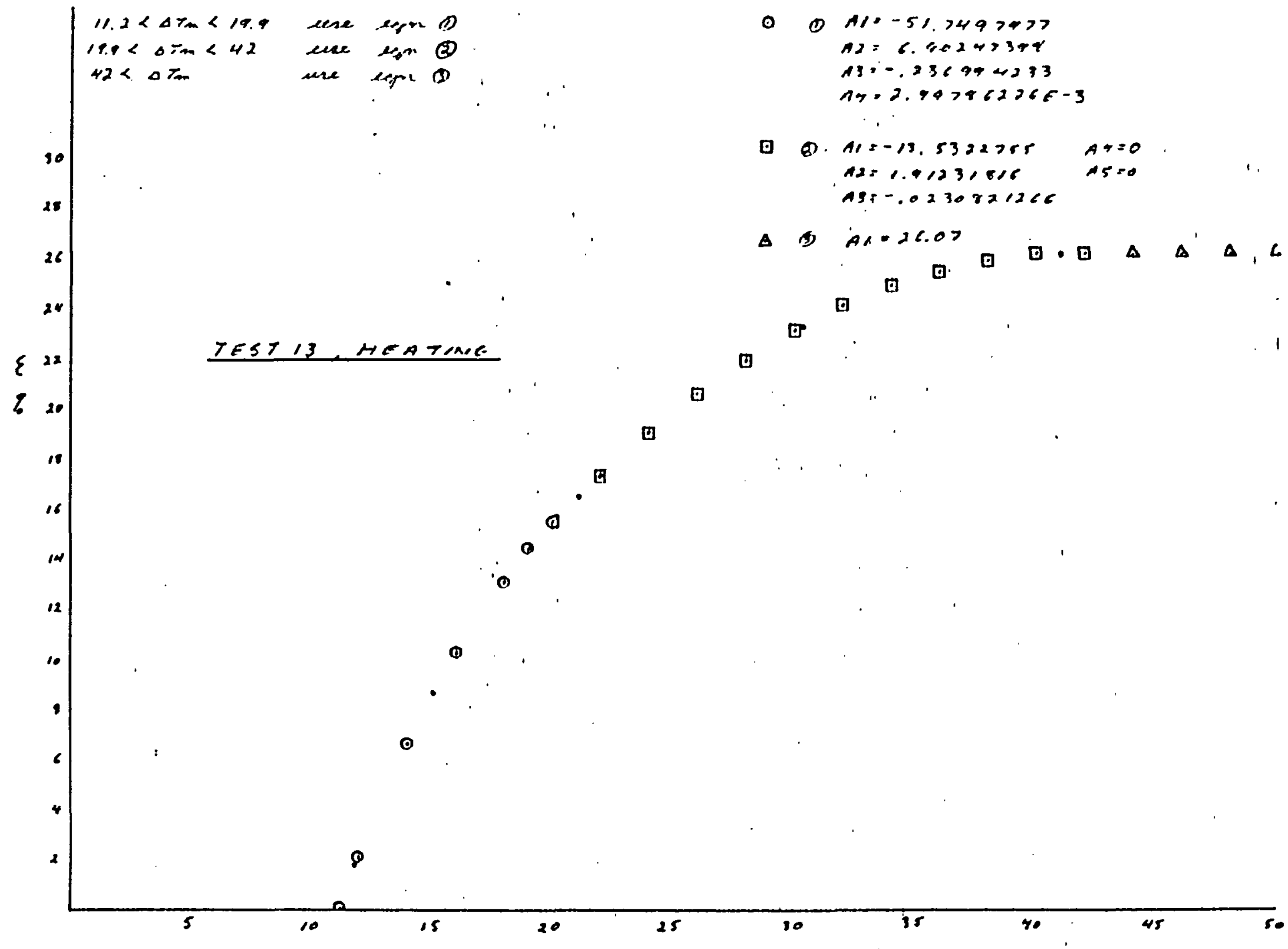
Y=A+BX+CX^2+DX^3+EX^4
WHERE A=-21.1305283
B=6.70695891
C=-.395398266
D=.0103823698
E=-9.92452029E-05

X	Y
0	-21.1305283
1	-14.8086845
2	-9.21673252
3	-4.29595085
4	9.99997382E-03
5	3.75507757
6	6.99085768
7	9.76653414
8	12.1289189
9	14.1224422
10	15.789152
11	17.1687147
12	18.2984148
13	19.2131548
14	19.9454553
15	20.5254552
16	20.9809113
17	21.3371986
18	21.6173102
19	21.8418574
20	22.0290695

TEST13 COOLING

Y=A+BX+CX^2+DX^3+EX^4
WHERE A=12.9743049
B=.629260694
C=-8.40843357E-03
D=0
E=0

X	Y
14	20.1359016
15	20.5213178
16	20.889917
17	21.2416994
18	21.5766649
19	21.8948136
20	22.1961454
21	22.4806603
22	22.7483583
23	22.9992395
24	23.2333038
25	23.4505513
26	23.6509818
27	23.8345956
28	24.0013924
29	24.1513724
30	24.2845355
31	24.4008818
32	24.5004111
33	24.5831237
34	24.6490193
35	24.6980981
36	24.73036
37	24.745805
38	24.7444332
39	24.7262445
40	24.6912389
41	24.6394165
42	24.5707772
43	24.4853211
44	24.3830481
45	24.2639582
46	24.1280514
47	23.9753278
48	23.8057873
49	23.6194299
50	23.4162557


11.2 < ΔTm < 19.9 use eqn ①
19.9 < ΔTm < 42 use eqn ②
42 < ΔTm use eqn ③
① A1 = -51.7497477
② A1 = -13.5322755
A4 = 0
A5 = 0
③ A1 = 26.07
TEST 13 , HEATING

EST13 HEATING

Y=A+BX+CX^2+DX^3+EX^4
HERE A=-51.7497977
B=6.90247398
C=-.236994233
D=2.99786226E-03
E=0

	Y
9	14.4046271
9.1	14.5183067
9.2	14.630682
9.3	14.741771
9.4	14.8515915
9.5	14.9601618
9.6	15.0674996
9.7	15.173623
9.8	15.2785501
9.9	15.3822987
20	15.4848868
20.1	15.5863325
20.2	15.6866538
20.3	15.7858686
20.4	15.8839948
20.5	15.9810506
20.6	16.0770539
20.7	16.1720226
20.8	16.2659748
20.9	16.3589285

TEST13 HEATING

Y=A+BX+CX^2+DX^3+EX^4
WHERE A=-13.5322755
B=1.91231816
C=-.0230821266
D=0
E=0

X	Y
19	14.4691218
19.1	14.5724108
19.2	14.675238
19.3	14.7776037
19.4	14.8795076
19.5	14.98095
19.6	15.0819307
19.7	15.1824497
19.8	15.2825072
19.9	15.3821029
20	15.4812371
20.1	15.5799096
20.2	15.6781204
20.3	15.7758696
20.4	15.8731572
20.5	15.9699831
20.6	16.0663474
20.7	16.16225
20.8	16.257691
20.9	16.3526704

[illegible]

[illegible]	[illegible]
11.57	1.83
[illegible]	[illegible]
[illegible]	[illegible]
[illegible]	[illegible]
40.92	[illegible]

TEST13 HEATING

Y=A+BX+CX^2+DX^3+EX^4
WHERE A=-51.7497977
B=6.90247398
C=-.236994233
D=2.99786226E-03
E=0

X	Y
10	-3.42661893
11	-.508731438
12	2.1330265
13	4.51664206
14	6.6601024
15	8.58139472
16	10.2985062
17	11.8294239
18	13.1921352
19	14.4046271
20	15.4848868
21	16.4509015
22	17.3206585
23	18.1121447

TEST13 HEATING

Y=A+BX+CX^2+DX^3+EX^4
WHERE A=-51.7497977
B=6.90247398
C=-.236994233
D=2.99786226E-03
E=0

X	Y
11	-.508731438
11.1	-.232426598
11.2	.0411349333
11.3	.311971143
11.4	.580100019
11.5	.845539548
11.6	1.10830772
11.7	1.36842251
11.8	1.62590193
11.9	1.88076394

TEST13 HEATING

Y=A+BX+CX^2+DX^3+EX^4
WHERE A=-13.5322755
B=1.91231816
C=-.0230821266
D=0
E=0

X	Y
17	12.3063986
18	13.4108424
19	14.4691218
20	15.4812371
21	16.447188
22	17.3669747
23	18.2405972
24	19.0680554
25	19.8493494
26	20.5844791
27	21.2734445
28	21.9162457
29	22.5128827
30	23.0633554
31	23.5676638
32	24.025808
33	24.4377879
34	24.8036036
35	25.123255
36	25.3967422
37	25.6240651
38	25.8052238
39	25.9402182
40	26.0290483
41	26.0717142
42	26.0682159
43	26.0185533
44	25.9227264
45	25.7807353
46	25.59258
47	25.3582604
48	25.0777765
49	24.7511284
50	24.378316
51	23.9593394
52	23.4941983
53	22.9828934
54	22.425424
55	21.8217903

```
]LIST

10  REM  THIS PROGRAM PREDICTS THE BEHAVIOUR OF 4 LOOP THERMOSIPHON GIVE
     N 1 LOOP BEHAVIOUR (TEST13A HEATING)
20  HOME
22  HTAB 10: INVERSE : PRINT "THERMOSIPHON HEAT EXCHANGER"
24  HTAB 11: PRINT "4 LOOP PREDICTION PROGRAM": PRINT : PRINT
25  NORMAL
26  INPUT "SIMULATE 2.3.OR 4 LOOPS? ":SL
27  IF SL < 2 OR SL > 4 THEN 26
28  PRINT
30  INPUT "FIND ONE POINT OR CALC. SERIES OF POINTS (O OR S):":D$
35  PRINT : INPUT "USE HEATING OR COOLING DATA FOR 1 LOOP  (H OR C):":H$

36  PRINT
40  PRINT "SELECT INITIAL DUCT TEMPS.(DEG.C)"
50  PRINT : INPUT "UPSTREAM EVAP:":KK
60  PRINT : INPUT "UPSTREAM COND: ":C1
70  IF D$ < > "S" THEN E1 = KK: GOTO 120
80  PRINT : PRINT : PRINT "SERIES OF POINTS"
90  INPUT "MAXIMUM EVAP. TEMP(DEG.C):":N1
100  INPUT "EVAP.TEMP.INCREMENT(DEG.C):":N2
103  PRINT : PRINT : INPUT "TO PRINT  RESULTS ON PRINTER TYPE P.ELSE
     PRESS RETURN:":P$
110  FOR E1 = KK TO N1 STEP N2
120 C4 = C1:C3 = C1:C2 = C1
130  GOSUB 500:LN = 1:EZ(1) = EF
140  GOSUB 600:LN = 2:EZ(2) = EF
150  IF Z = 0 THEN  GOSUB 1500
160  IF Z > = 1 THEN 185
170  GOSUB 1710
180  GOTO 130
185  IF SL = 2 THEN 300
190  GOSUB 700:LN = 3:EZ(3) = EF
200 LL = LL + 1: IF LL = 1 THEN BC = 0
210  IF Z = 1 THEN  GOSUB 1500
220  GOSUB 1710
230  IF Z = 1 THEN 130
240 DD = DD + 1: IF DD = 1 THEN BC = 0
245  IF SL = 3 THEN 300
250  GOSUB 800:LN = 4:EZ(4) = EF
260  IF Z = 2 THEN  GOSUB 1500
270  GOSUB 1710
280  IF Z = 3 THEN 300
290  GOTO 130
300  GOSUB 2000
310  IF D$ = "S" THEN 330
320  GOTO 2260
330 LL = 0:DD = 0:Z = 0:BC = 0: NEXT E1
340  GOTO 2260
500  REM  LOOP 1 CALC
510 TM(1) = E1 - C4
520 AA = TM(1): GOSUB 1000
530 TE(1) = EF / 100 * TM(1)
540 E2 = E1 - TE(1):C5 = C4 + TE(1)
550 E2 =  INT (E2 * 1000 + .5) / 1000:C5 =  INT (C5 * 1000 + .5) / 1000
560  RETURN
600  REM  SECOND LOOP
610 TM(2) = E2 - C3
620 AA = TM(2): GOSUB 1000
630 TE(2) = EF / 100 * TM(2)
640 C4 = C3 + TE(2):E3 = E2 - TE(2)
650 C4 =  INT (C4 * 1000 + .5) / 1000
660 E3 =  INT (E3 * 1000 + .5) / 1000
670  RETURN
700  REM  THIRD LOOP
710 TM(3) = E3 - C2
720 AA = TM(3): GOSUB 1000
730 TE(3) = EF / 100 * TM(3)
740 C3 = C2 + TE(3):E4 = E3 - TE(3)
750 C3 =  INT (C3 * 1000 + .5) / 1000
760 E4 =  INT (E4 * 1000 + .5) / 1000
770  RETURN
800  REM  FOURTH LOOP
810 TM(4) = E4 - C1
820 AA = TM(4): GOSUB 1000
830 TE(4) = EF / 100 * TM(4)
840 C2 = C1 + TE(4):E5 = E4 - TE(4)
850 C2 =  INT (C2 * 1000 + .5) / 1000
860 E5 =  INT (E5 * 1000 + .5) / 1000
870  RETURN
1000  REM  EFFECTIVENESS CALC.
1010  IF H$ = "C" THEN 1100
1020  IF AA < 11.2 THEN EF = 0: GOTO 1160
1030  IF AA > 19.9 THEN 1060
1040 A1 =  - 51.7497977:A2 = 6.90247398:A3 =  - .236994233:A4 = 2.997862
      26E - 3:A5 = 0
1050  GOTO 1140
1060 A1 =  - 13.5322755:A2 = 1.91231816:A3 =  - .0230821266:A4 = 0:A5 =
      0
1070  GOTO 1140
1100  IF AA < 4.9 THEN EF = 0: GOTO 1160
1110  IF AA > 15 THEN 1130
1120 A1 =  - 29.9316819:A2 = 8.23475687:A3 =  - .490468871:A4 = .0128959
      49:A5 =  - 1.23075172E - 4: GOTO 1140
1130 A1 = 12.9743049:A2 = .629260694:A3 =  - 8.40843357E - 3:A4 = 0:A5 =
      0
```

```
1140 EF = A1 + A2 * AA + A3 * AA ^ 2 + A4 * AA ^ 3 + A5 * AA ^ 4
1150 EF = INT (EF * 1000 + .5) / 1000
1160  RETURN
1500  REM
1510 BB = EZ(LN): IF  ABS (BB - BC) < .01 THEN Z = Z + 1
1520 BC = EZ(LN)
1530  RETURN
1700  REM
1710  HOME : PRINT : PRINT : PRINT : PRINT : PRINT : PRINT : PRINT : PRINT
      TAB( 12):Z + 2:"  LOOP CALCULATIONS"
1720  PRINT  TAB( 16):"PLEASE WAIT"
1730  RETURN
2000 -REM  PRINT ROUTINE
2010  IF P$ = "P" THEN  PR# 1: PRINT  CHR$ (27): CHR$ (77): CHR$ (15)
2020  HOME
2022  IF SL = 2 THEN E5 = E3:E4 = E3
2024  IF SL = 3 THEN E5 = E4
2025  PRINT : HTAB 10: PRINT "PREDICTION FOR ":SL:" LOOPS WORKING"
2030  PRINT : PRINT "AIR TEMPS. THROUGH COILS (DEG.C)"
2040  PRINT "1-UPSTREAM:  5-DOWNSTREAM": PRINT
2050  PRINT "EVAP.  E1     E2     E3     E4     E5"
2060  HTAB 7: PRINT E1:: HTAB 14: PRINT E2:: HTAB 21: PRINT E3:: HTAB 28
     : PRINT E4:: HTAB 35: PRINT E5
2070  PRINT : PRINT "COND.  C1     C2     C3     C4     C5"
2080  HTAB 7: PRINT C1:: HTAB 14: PRINT C2:: HTAB 21: PRINT C3:: HTAB 28
     : PRINT C4:: HTAB 35: PRINT C5
2090  PRINT : PRINT
2100  PRINT "LOOP# EFFECT.  TE(I)   TM(I)"
2110  FOR IJ = 1 TO 4
2120 TM(IJ) =  INT (TM(IJ) * 1000 + .5) / 1000
2130 TE(IJ) =  INT (TE(IJ) * 1000 + .5) / 1000
2140  PRINT "  ":IJ:: HTAB 8: PRINT EZ(IJ):: HTAB 15: PRINT TE(IJ):: HTAB
     23: PRINT TM(IJ)
2150  NEXT IJ
2160 LE = (E1 - E5) / (E1 - C1) * 100
2165 LE =  INT (LE * 1000 + .5) / 1000
2170  PRINT : PRINT : PRINT "TOTAL LOOP EFFECT. =":LE:" %"
2180  PRINT "MAX.TEMP.DIFF.(DEG.C)= ":E1 - C1
2185  PRINT : PRINT
2190  PR# 0
2200  IF D$ = "S" THEN 2240
2210  PRINT : PRINT : INPUT " TO PRINT RESULTS ON PRINTER TYPE P.    ELS
     E TYPE RETURN:":P$
2220  IF P$ = "P" THEN 2000
2230  GOTO 2250
2240  FOR FF = 1 TO 500: NEXT FF
2250  RETURN
2260  END
```

VITA AUCTORIS

1959	Born in Welland, Ontario on March 20
1978	Completed Secondary School Honours program from W.A.Porter Collegiate, Scarborough, Ontario
1983	Completed the Bachelor of Applied Science Degree in Mechanical Engineering from the University of Windsor, Windsor, Ontario
1985	Currently a candidate for the degree of Master`s of Applied Science, and have accepted an energy related position with Union Gas of Chatham, Ontario